US006250807B1

(12) United States Patent
Mori et al.

(10) **Patent No.: US 6,250,807 B1**
(45) **Date of Patent: Jun. 26, 2001**

(54) **HYDRODYNAMIC TYPE BEARING AND HYDRODYNAMIC TYPE BEARING UNIT**

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(*) Notice: Subject to any disclaimer, the term of this patent is extended or adjusted under 35 U.S.C. 154(b) by 0 days.

(21) Appl. No.: **09/391,594**

(22) Filed: **Sep. 8, 1999**

(30) **Foreign Application Priority Data**

Sep. 10, 1998 (JP) 10-257116
Apr. 16, 1999 (JP) 11-110017

(51) **Int. Cl.**[7] **F16C 32/06**
(52) **U.S. Cl.** **384/100**; 384/107
(58) **Field of Search** 384/100, 107, 384/111, 112, 113, 114, 121, 123

(56) **References Cited**

U.S. PATENT DOCUMENTS

3,445,148 * 5/1969 Harris et al. 384/279
5,127,744 * 7/1992 White et al. 384/112
5,357,162 * 10/1994 Aiyoshizawa et al. .... 384/112 X
5,683,183 * 11/1997 Tanaka et al. 384/100
5,704,718 * 1/1998 Mori et al. 384/279
5,810,479 * 9/1998 Miyasaka et al. 384/107
5,941,646 * 8/1999 Mori et al. 384/279

* cited by examiner

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(74) *Attorney, Agent, or Firm*—Arent Fox Kintner Plotkin & Kahn, PLLC

(57) **ABSTRACT**

A thrust bearing section (**14**) is constituted by one bearing end face (**11*f*1**) of a hydrodynamic type oil-impregnated sintered bearing (**11**) and a flange portion (**13*a***) provided on a rotating shaft (**13**). The squareness between the bearing end face (**11*f*1**) and the bearing inner periphery (**11*h***) is set within 3 μm, and the squareness between the flange portion (**13*a***) and the outer periphery of the rotating shaft (**13**) is set within 2 μm. The bearing bore diameter d and the bearing length L of the hydrodynamic type oil-impregnated sintered bearing are set as L≦1.2 d, and a radial bearing surface (**11*b***) is arranged at one place on the bearing inner periphery (**11*h***).

13 Claims, 16 Drawing Sheets

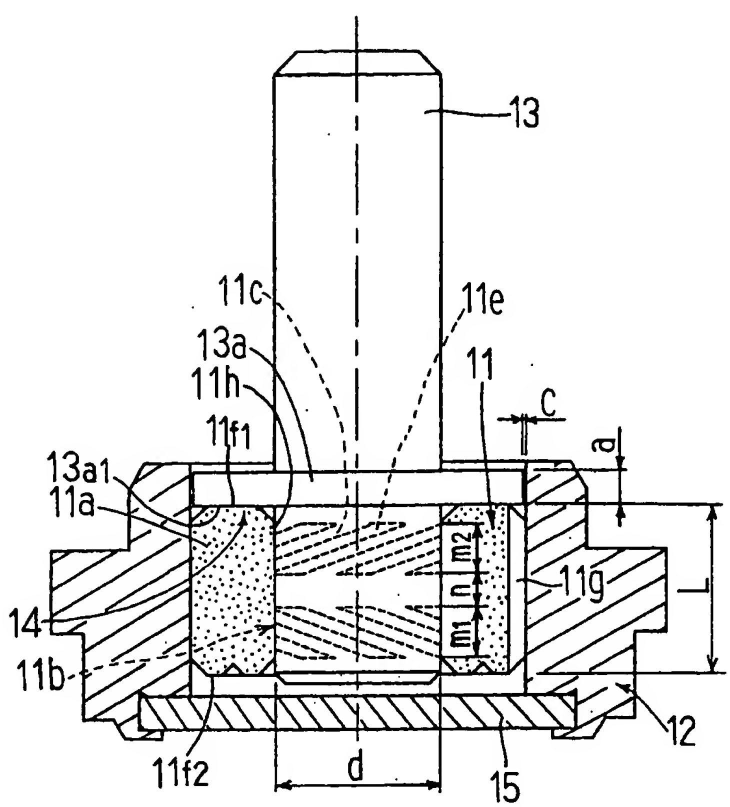
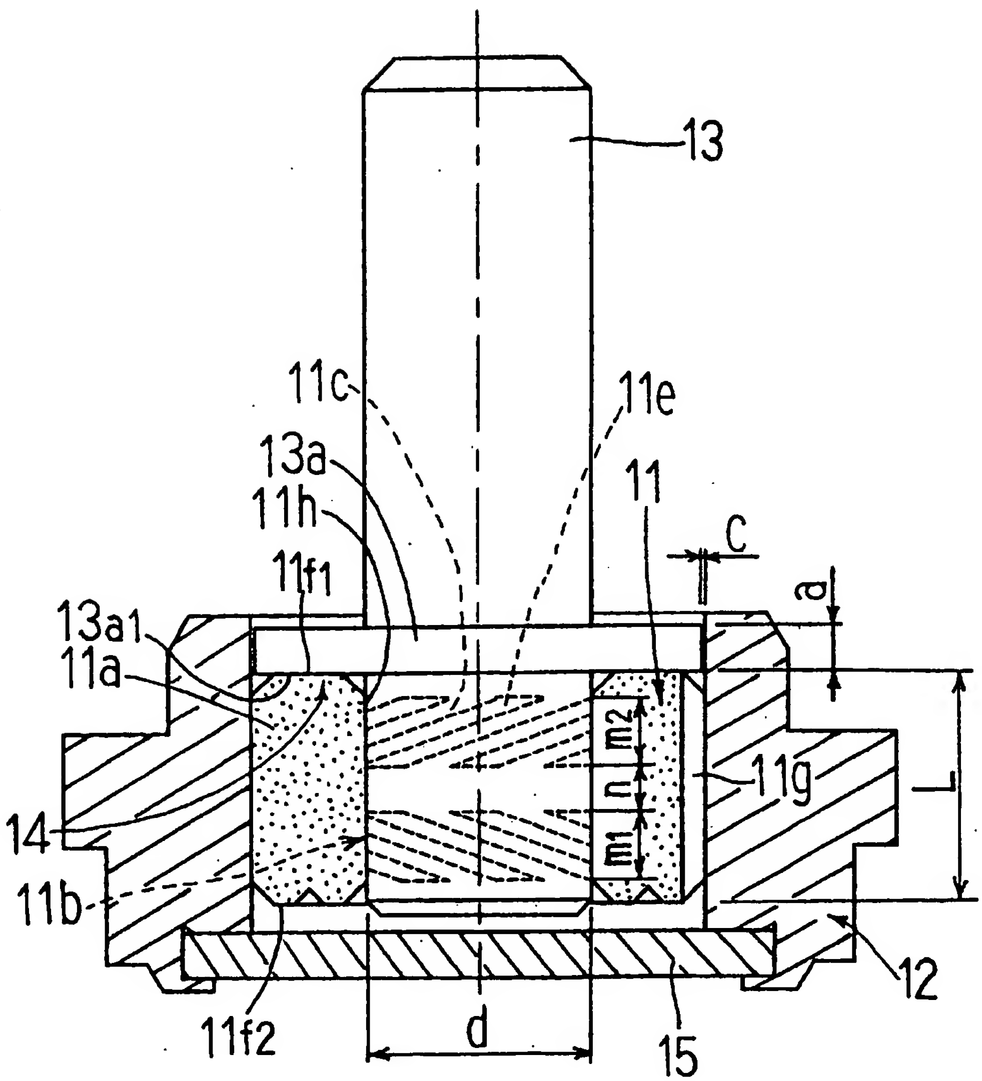


FIG.1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13a
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11h
c
11f1
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m2
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11g
L
14
m1
11b
12
11f2
d
15

FIG.2(A)

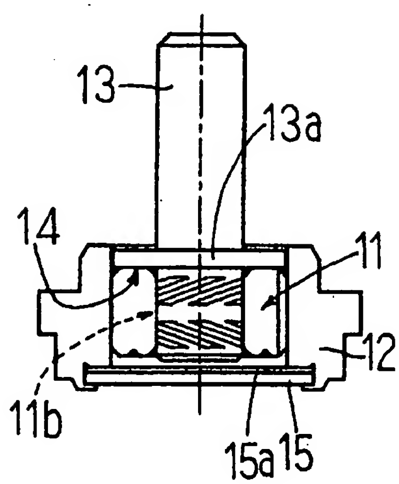


FIG.2(B)

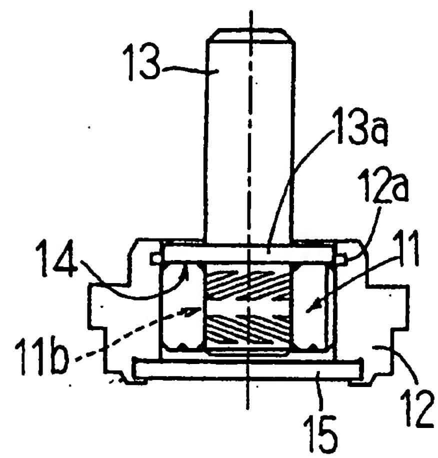


FIG.3(A)

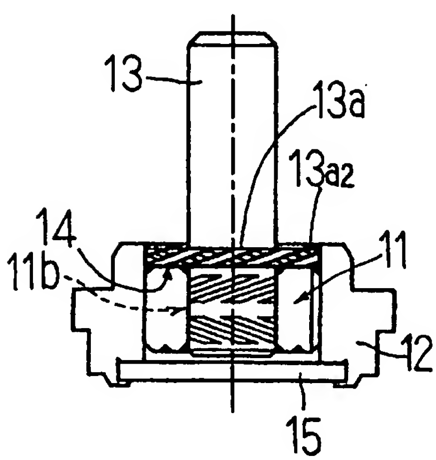


FIG.3(B)

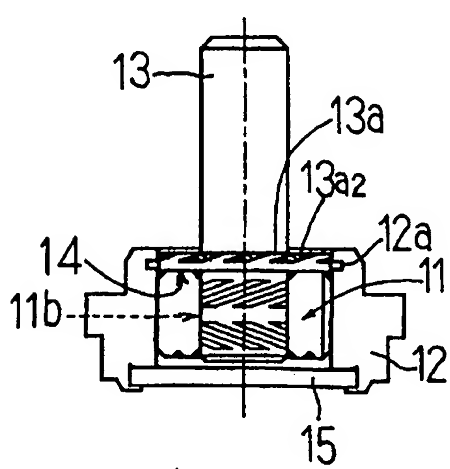
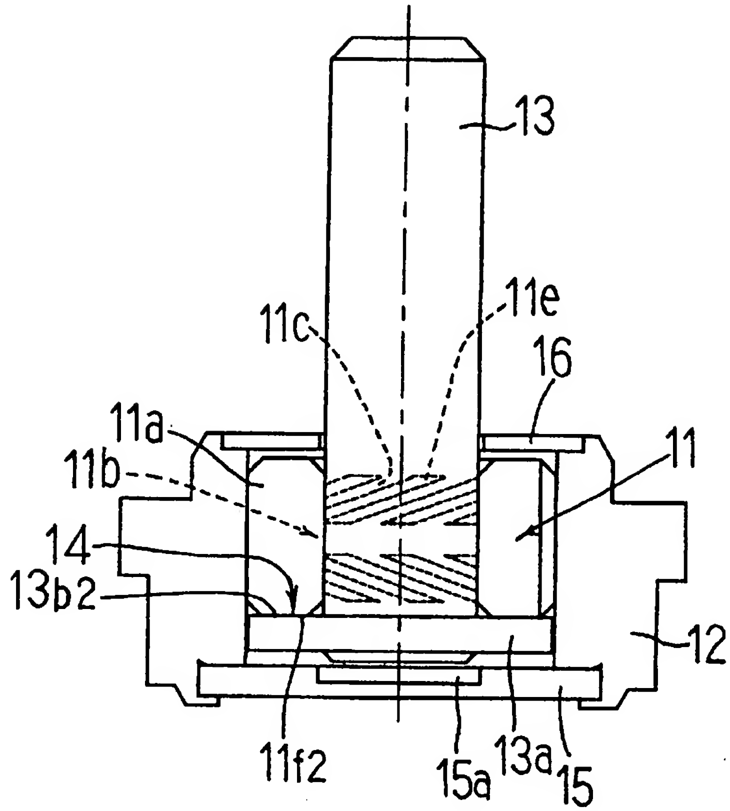


FIG.4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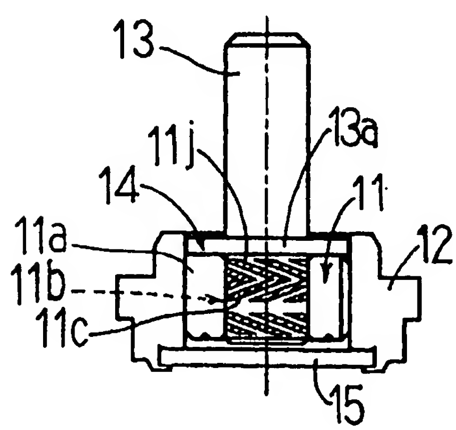
11c
11e
16
11a
11b
11
14
13b2
12
11f2
15a
13a
15

FIG.5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13a
11j
14
11
11a
12
11b
11c
15

FIG.6(A)

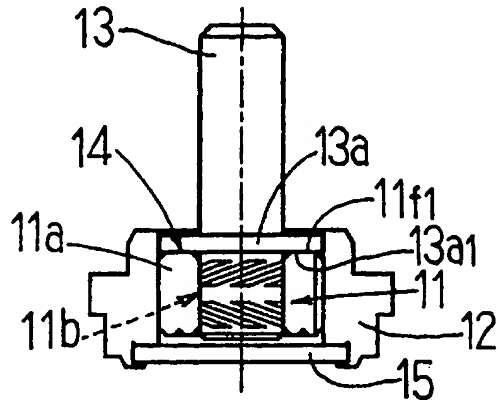


FIG.6(B)

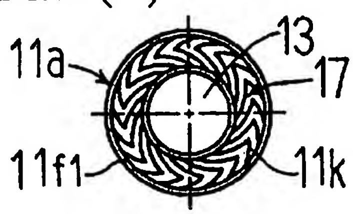
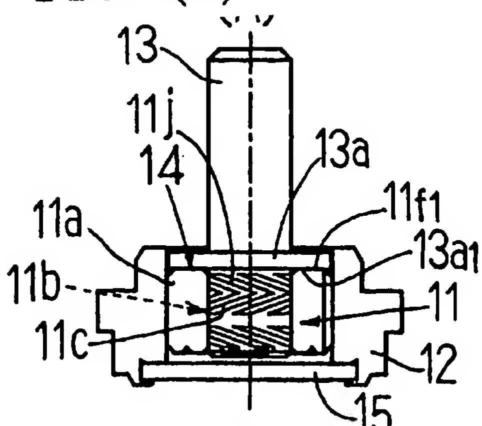


FIG.7(A)



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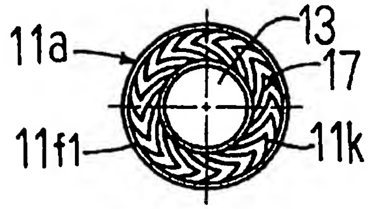

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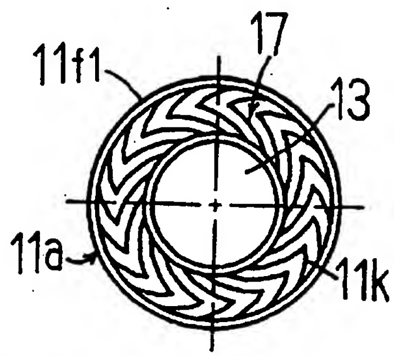


FIG.8(A)

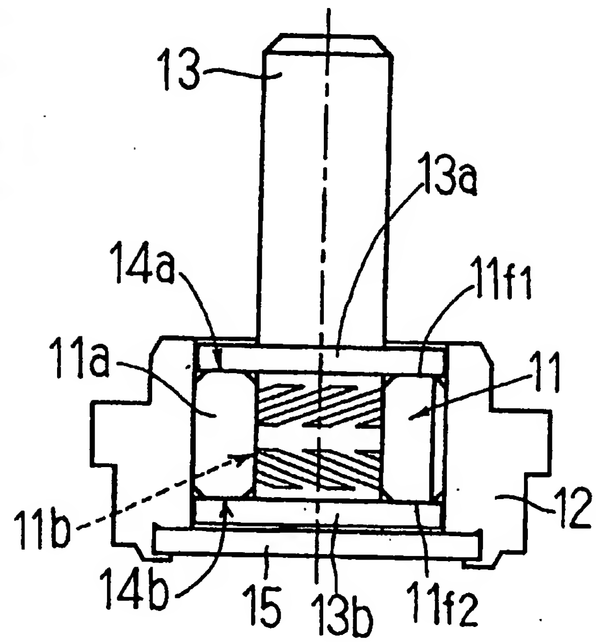


FIG8(C)

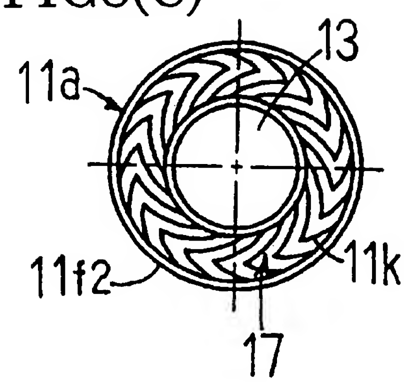


FIG.9(B)

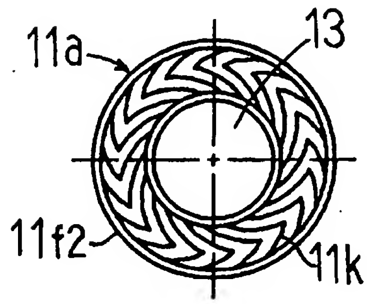


FIG.9(A)

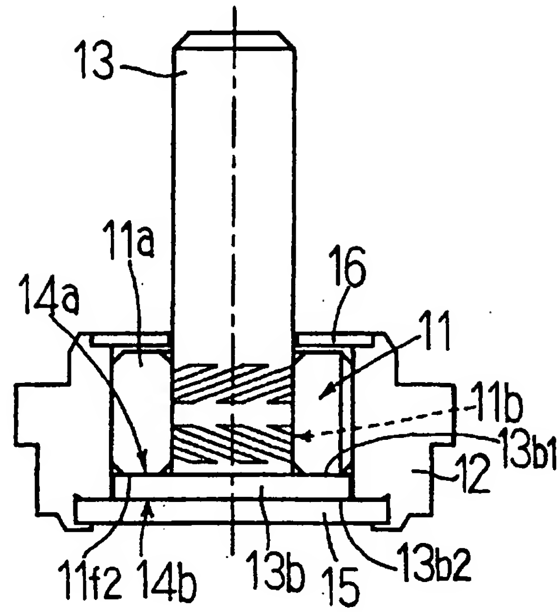


FIG.9(C)

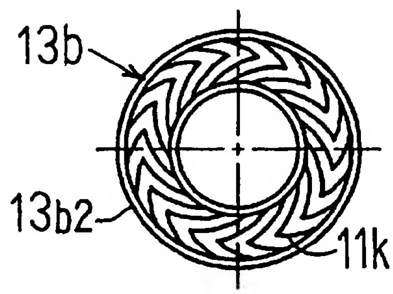


FIG.10

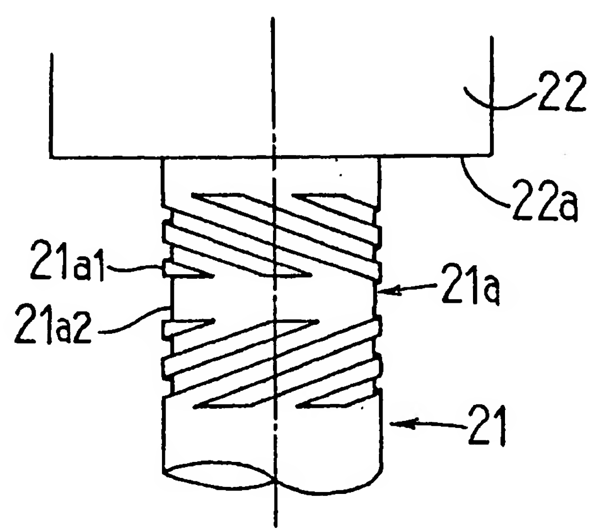


FIG.11

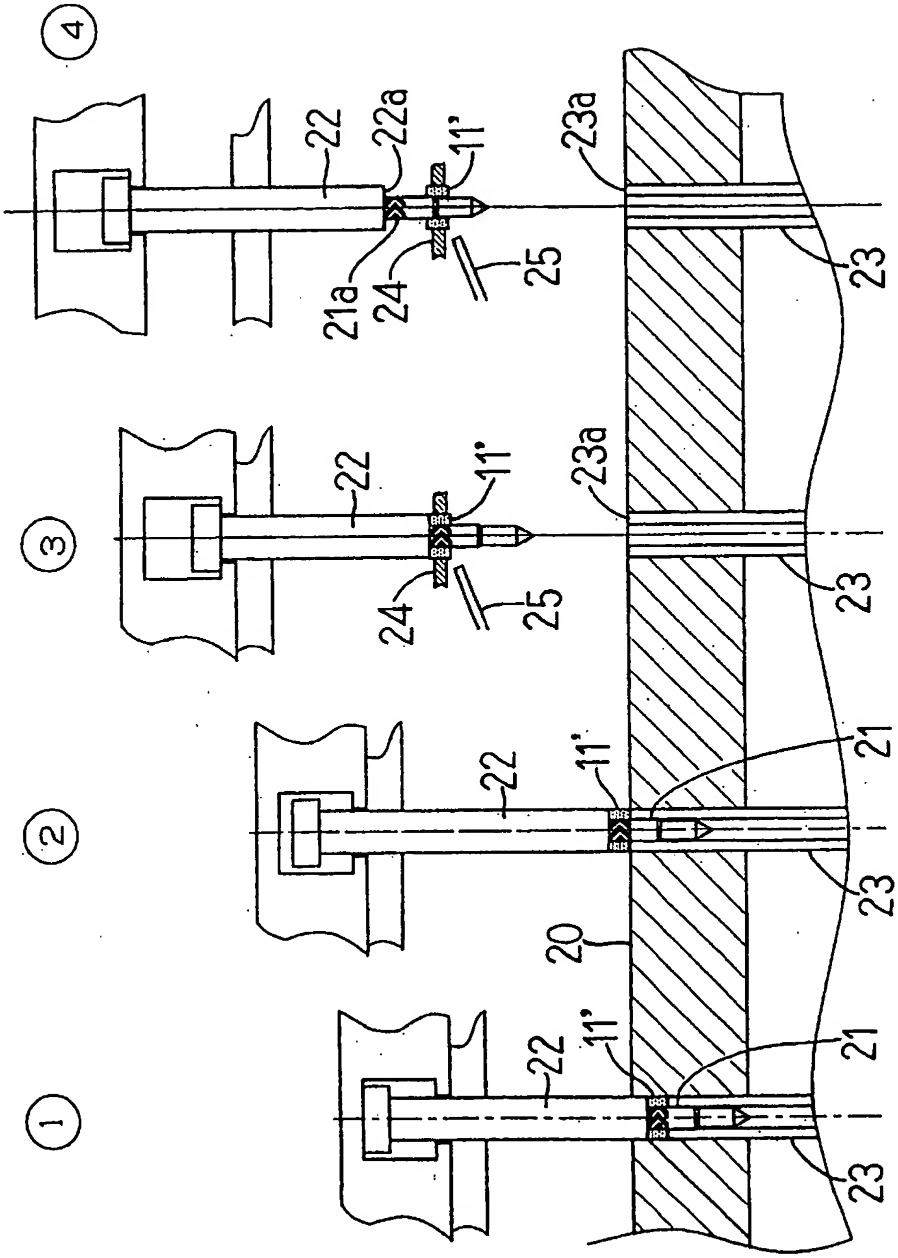


FIG.12
(PRIOR ART)

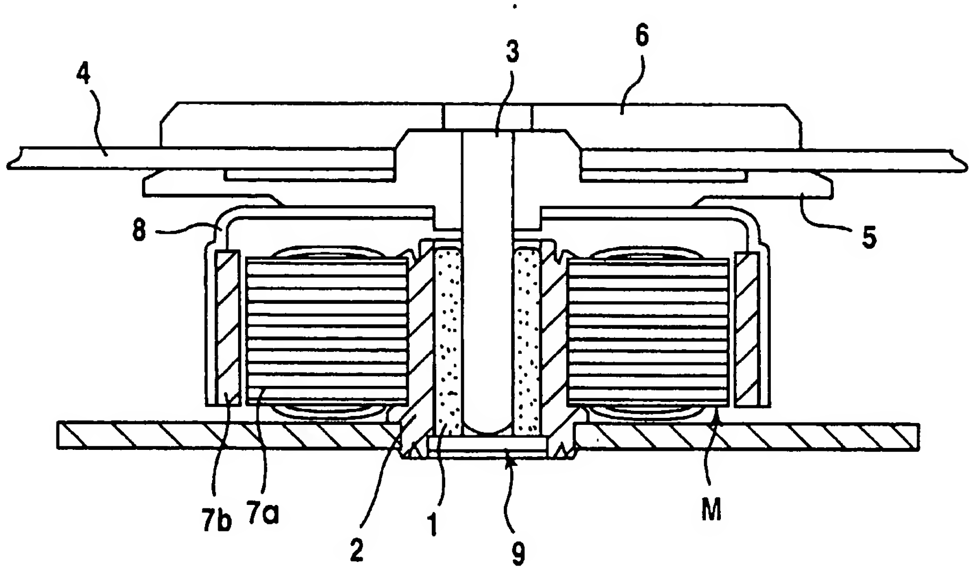


FIG.13
(PRIOR ART)

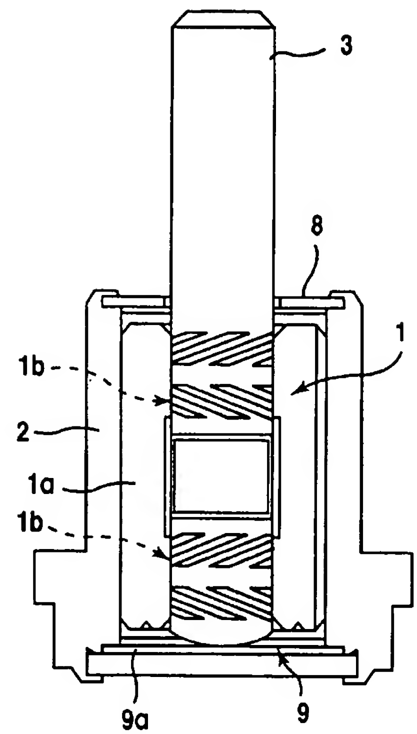


FIG.14
(PRIOR ART)

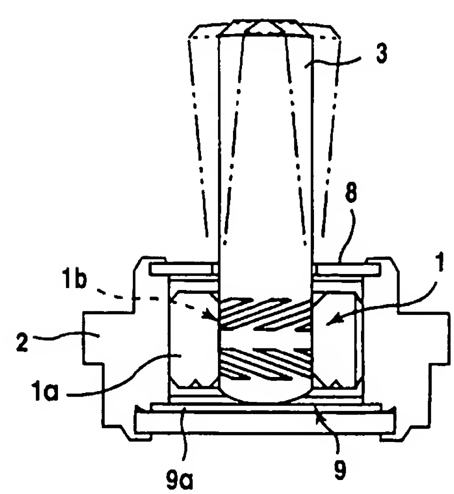


FIG.15

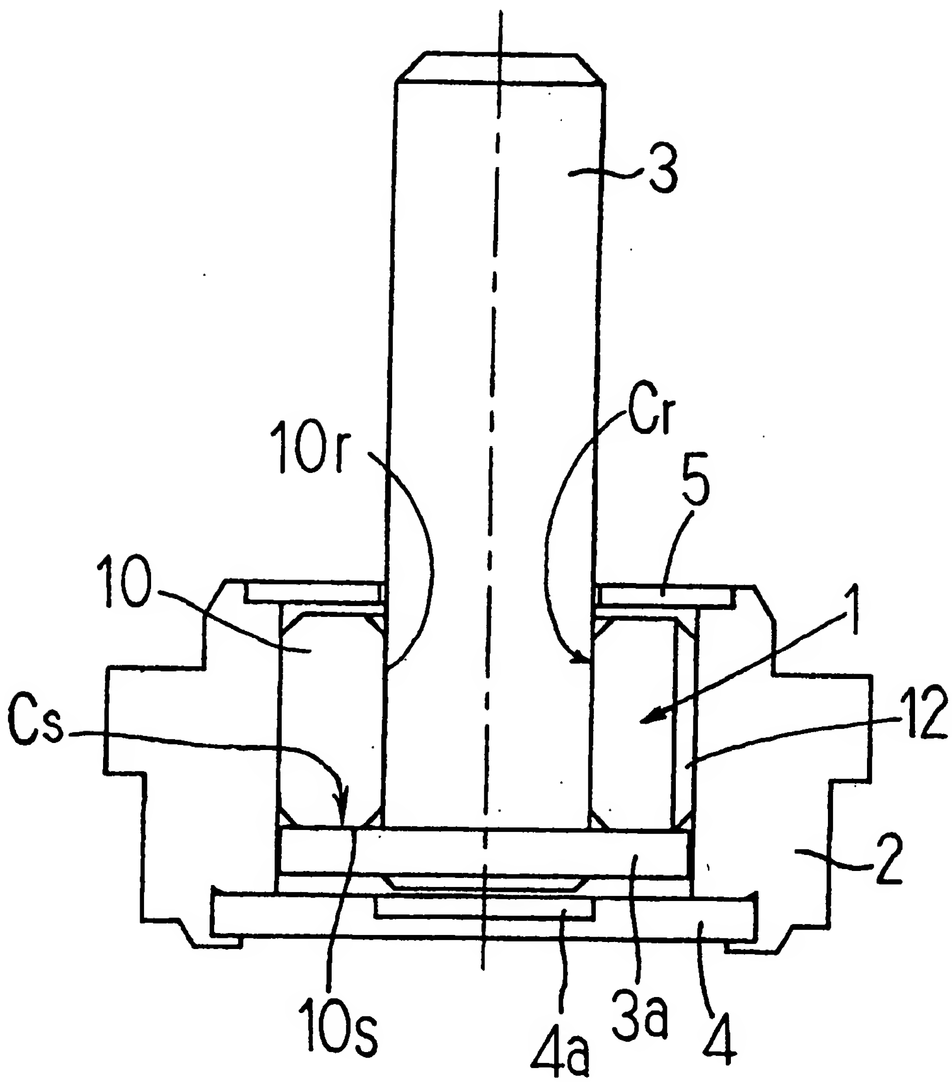


FIG.16(A)

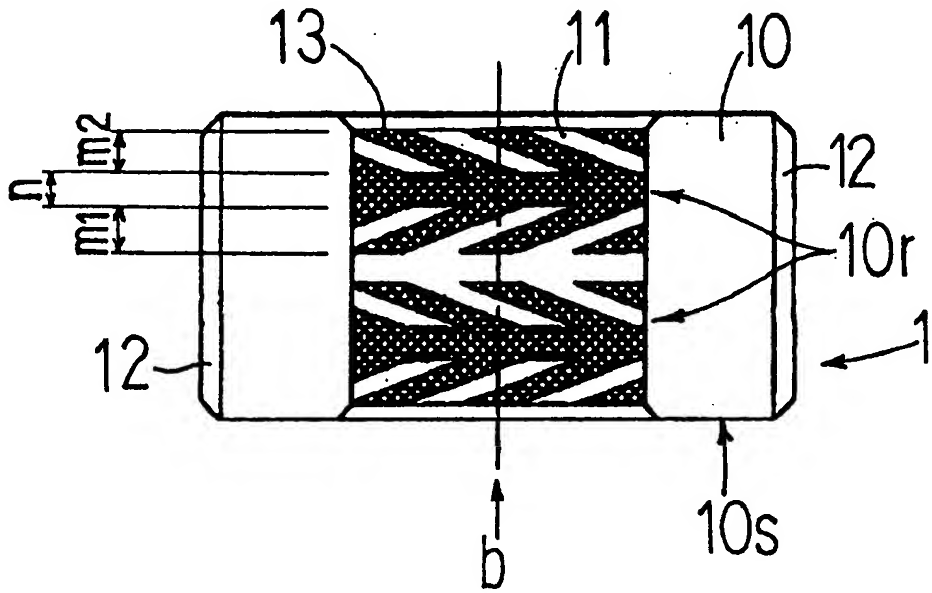


FIG.16(B)

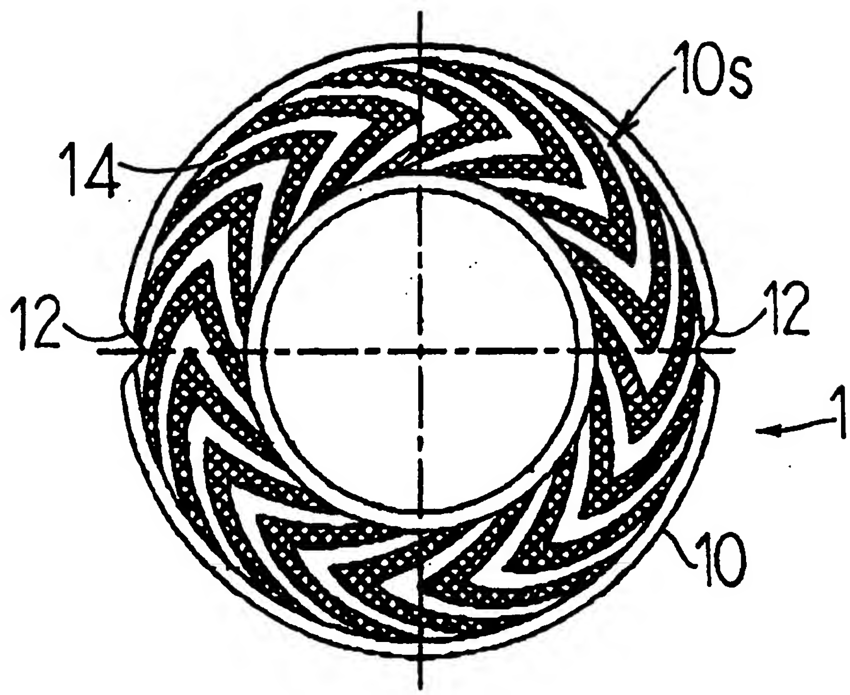


FIG.17

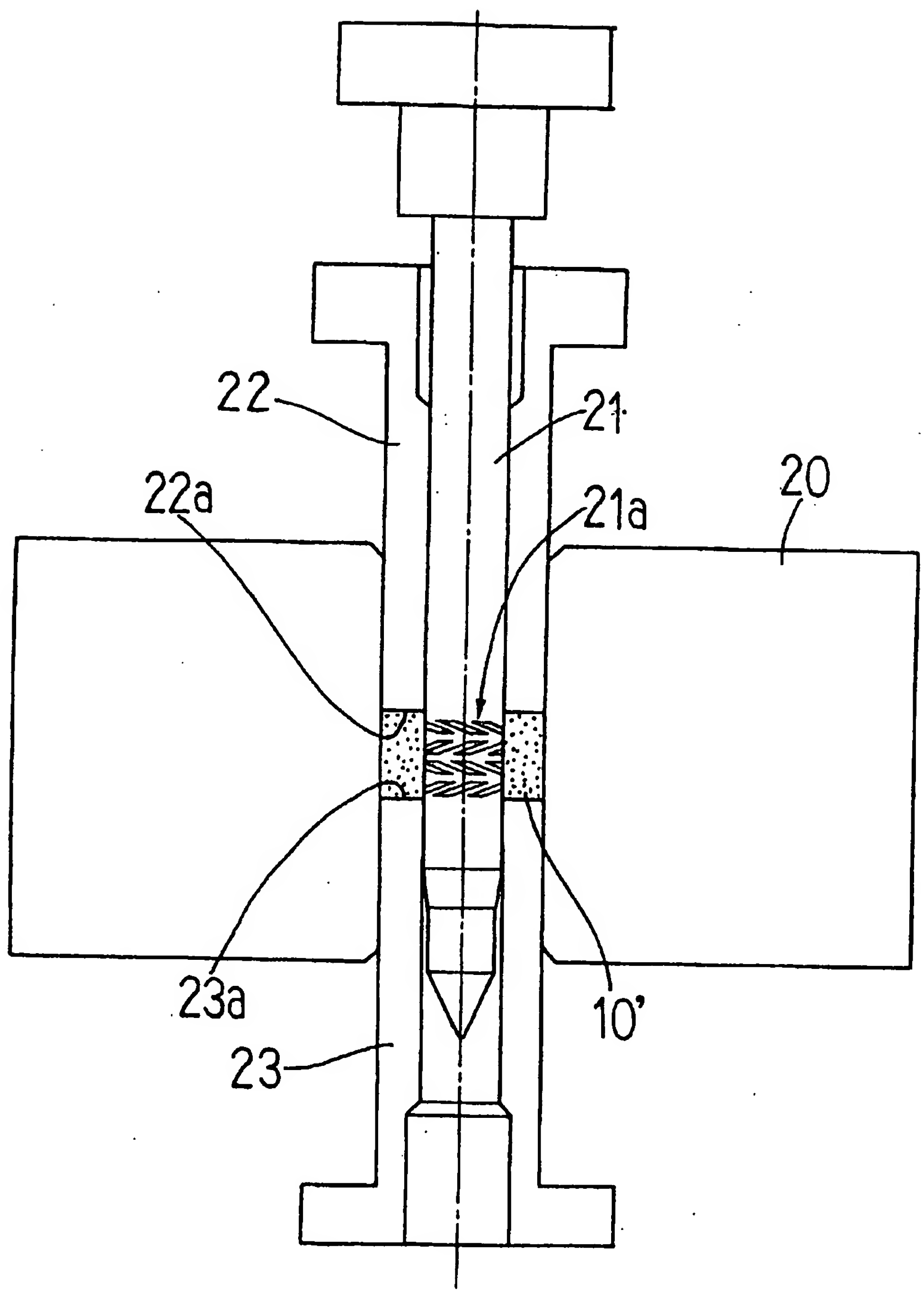


FIG.18

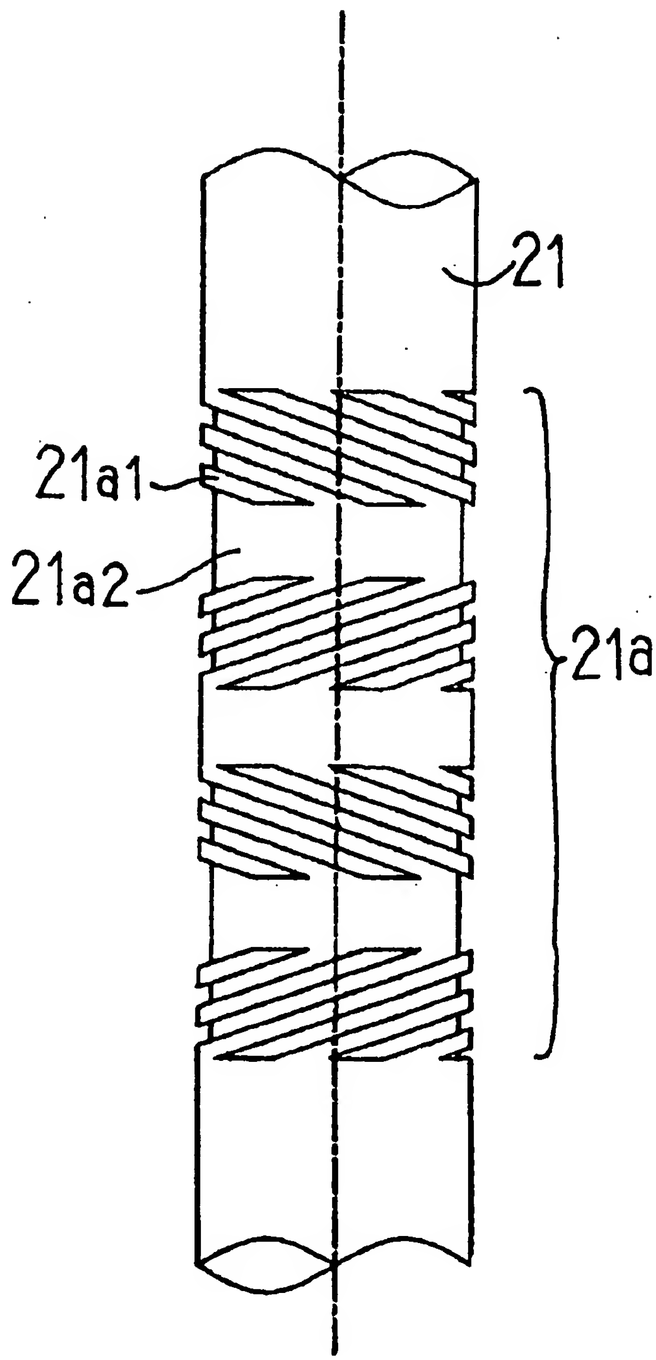


FIG.19

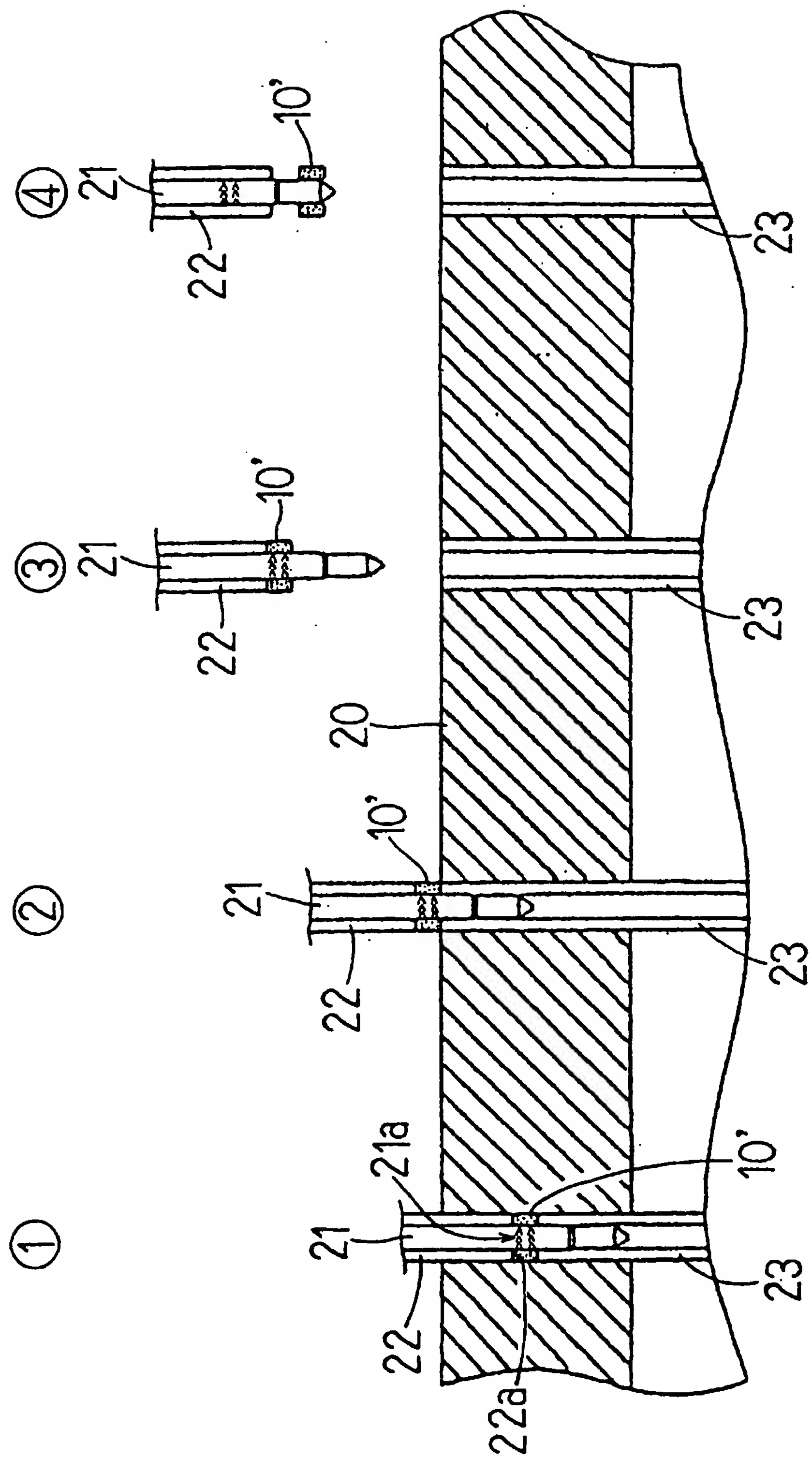
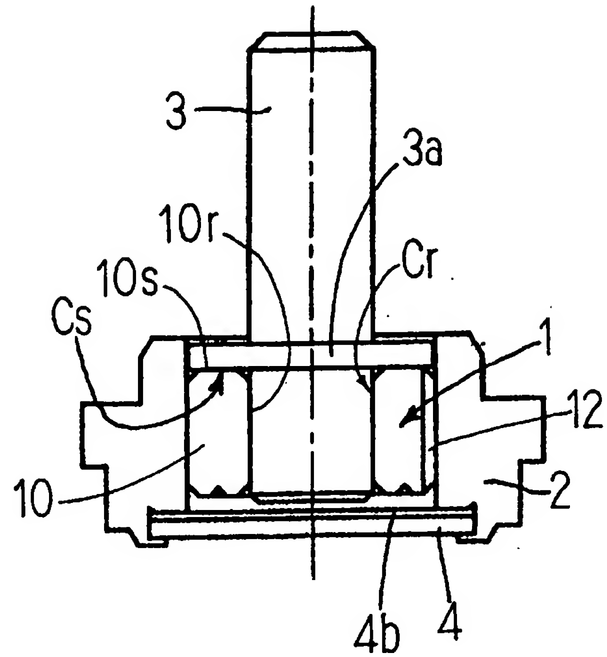


FIG.20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10s
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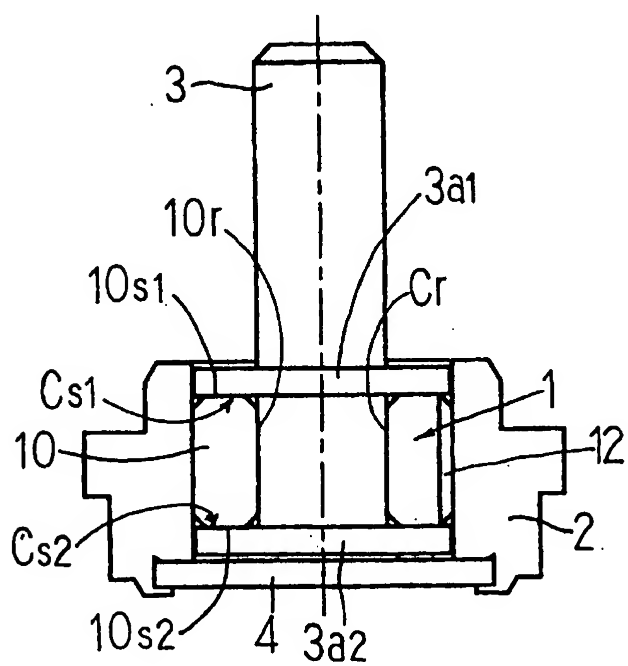
2
4b
4

FIG.21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Cr
Cs1
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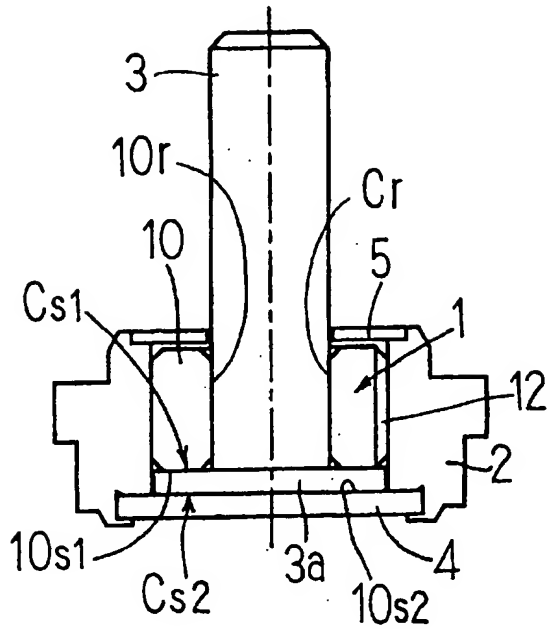
Cs2
2
10s2
4
3a2

FIG.22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Cs1
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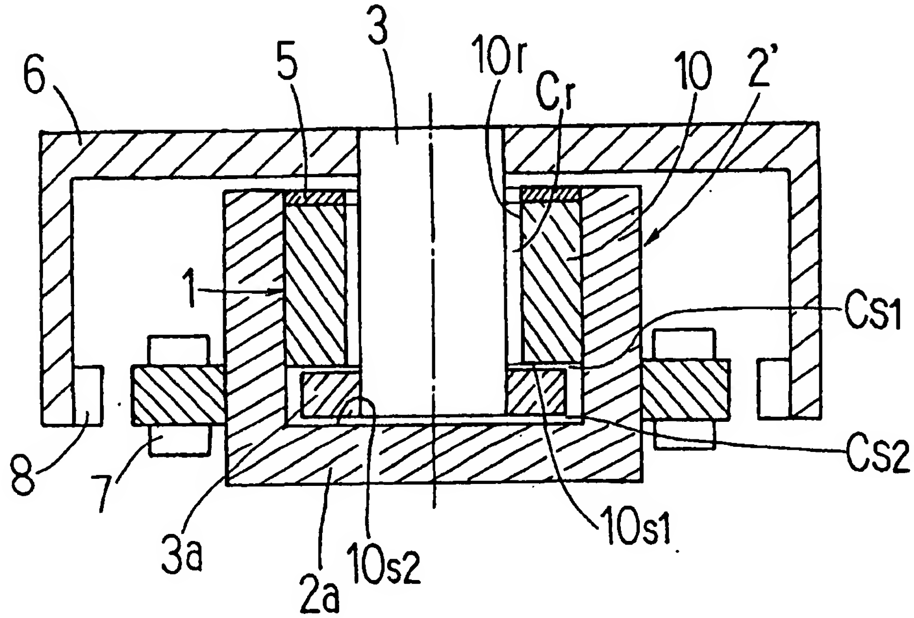
10s1
Cs2
3a
10s2
4

FIG.23


6
5
3
10r
Cr
10
2'
1
Cs1
Cs2
8
7
3a
2a
10s2
10s1

HYDRODYNAMIC TYPE BEARING AND HYDRODYNAMIC TYPE BEARING UNIT

BACKGROUND OF THE INVENTION

1. Field of the Invention

The invention relates to a hydrodynamic type bearing, particularly a hydrodynamic type oil-impregnated sintered bearing, and to a hydrodynamic type bearing unit employing the same. Hydrodynamic type oil-impregnated sintered bearings are especially suited to bearings in use for information equipment, i.e., bearings for disc drives in optical disc devices including DVD-ROMs and DVD-RAMs, magneto-optical disc devices including MOs, and magnetic disc devices including HDDs and high capacity floppy disc drives (FDDS) such as HiFDs and Zips, or bearings for polygon scanner motors in LBPs and the like. In particular, hydrodynamic type oil-impregnated sintered bearings are suitably applied to bearings in thinner models of motors.

2. Description of the Prior Art

Spindle motors for the information equipment mentioned above are required for further improvements in high-speed rotational accuracy, higher speeds, lower costs, lower noise, and the like. One of the component parts determining these performance requirements is a bearing for supporting the spindle of a motor. In recent years, studies have been made on the use of a hydrodynamic type bearing, especially of a so-called hydrodynamic type oil-impregnated sintered bearing, as such bearing. In a hydrodynamic type oil-impregnated sintered bearing, the bearing body of sintered metal is impregnated with lubricating oil or lubricating grease, and a lubricating film is formed in a bearing clearance by means of the hydrodynamic action of hydrodynamic pressure generating grooves provided in the bearing surface, so as to support a spindle without contact. This hydrodynamic type oil-impregnated sintered bearing, having the features of high rotational accuracy, low noise and the like despite of its low costs, appears to well meet the aforesaid performance requirements.

FIG. 12 shows an example of a spindle motor in an optical disc device, employing a hydrodynamic type oil-impregnated sintered bearing **1**. As shown in the figure, this spindle motor comprises the hydrodynamic type oil-impregnated sintered bearing **1**, a housing **2** for containing the bearing **1**, a rotating shaft **3** supported by the bearing **1**, a turntable **5** and a damper **6** for supporting and fixing an optical disc **4**, and a motor section M composed of a stator **7a** and a rotor **7b**. The spindle motor is configured so that energizing the stator **7a** brings a rotor case **8** integrated with the rotor **7b**, the turntable **5**, the optical disc **4**, and the damper **6** into integral rotation.

The hydrodynamic type oil-impregnated sintered bearing **1** is composed of a porous bearing body formed in a thick cylindrical shape, and oil stored in the pores of the bearing body by means of the impregnation with lubricating oil or lubricating grease. In the inner periphery of the bearing body, a pair of bearing surfaces opposed to the outer periphery of the rotating shaft via bearing clearances are formed so as to be axially separated from each other. In each bearing surface are formed hydrodynamic pressure generating grooves slanting against an axial direction.

As shown in FIGS. **12** and **13**, a thrust load on the rotating shaft **3** is supported by a thrust bearing **9** arranged at the bottom of the housing **2**. The thrust bearing **9** typically has a configuration (so-called a pivot bearing) in which the spherical shaft end thereof slides on a resin washer **9a** having high lubricity provided at the bottom of the housing **2**.

The pivot bearing, however, may suffer a change in shaft position with lapse of time due to a recess in the washer **9a** created by elastic deformation, plastic deformation, deformation from friction, and the like of the washer **9a**. The change in shaft position cause variations in disc position in the cases of HDD devices, and variations in mirror position in the cases of polygon scanner motors in LBPS, greatly affecting the motor performance. As measures against this, the washer **9a** could be formed of metal material or ceramic material; in such case, however, the shaft will be worn out to turn the spherical surface of the shaft end into a flat surface, possibly producing the problems of a change in shaft position, a rise in torque, fluctuations in torque, and the like.

Moreover, in recent years, the spindle motors are often required for thinner models in view of the mounting of optical disc devices and HDD devices on notebook type computers and the like, while the configuration that the bearing surfaces **1b** are arranged axially at two places as described above has a limit in obtaining thinner models. As shown in FIG. **14** for example, a thinner model can be obtained by arranging the bearing surface **1b** at only one place. This produces, however, the problem of a decrease in rigidity with respect to moment loads. In other words, since the rotating shaft **3** at the portion projecting from the bearing **1** is subjected to eccentric loads from the rotor case **8** having the rotor magnet **7b** fixed thereto, the disc **4**, the turntable **5**, the clamper **6**, and the like, it is feared that the accuracy in shaft run-out might be deteriorated by the moment loads.

Such hydrodynamic type bearing has hydrodynamic pressure generating grooves of herringbone type, spiral type, or the like for generating a hydrodynamic pressure formed in the inner periphery (radial bearing surface) of its almost-cylindrical-shaped sleeve material. A conventional method for forming hydrodynamic pressure generating grooves is known in which a rod-shaped jig, holding a plurality of balls harder than the bearing material arranged circumferentially at equal intervals, is inserted into the inner periphery of the bearing material, and the jig is rotated and fed to put the balls into spiral movements while pressing the balls against the inner periphery of the material to form by rolling (plastic working) the hydrodynamic pressure generating grooves (Japanese Patent No. 2541208).

In such hydrodynamic type bearing, a thrust bearing surface having hydrodynamic pressure generating grooves is sometimes provided on an end face of the bearing or a surface opposed thereto of the spindle, in order to non-contact support the spindle in a thrust direction. These hydrodynamic pressure generating grooves in the thrust bearing surface are typically formed by pressing.

The above-described rolling of hydrodynamic pressure generating grooves, however, creates heaving at regions adjacent to the hydrodynamic pressure generating grooves in working. The heaving must be removed by a lathe or a reamer (Japanese Patent Laid-Open Publication No.Hei 8-232958), complicating the processes. Besides, in the removing, positioning need to be performed with end faces of the bearing pressed against the jig; therefore, the end faces of the bearing must have been finished with a high degree of accuracy, and the accuracy of the end faces should be maintained in the removing as well, making the work troublesome.

Moreover, since the thrust bearing surface is worked in a separate process from that of the radial bearing surface, a bearing surface formed in the preceding process may decrease in accuracy during the following process, producing a difficulty in quality control.

SUMMARY OF THE INVENTION

In view of the foregoing, an object of the present invention is-to provide a hydrodynamic type bearing, particularly a hydrodynamic type oil-impregnated sintered bearing, being capable of maintaining a desired bearing performance for a long period of time and realizing thinner models without producing a decrease in the accuracy of shaft run-out and the like.

Another object of the present invention is to simplify the fabrication processes of a hydrodynamic type bearing having a radial bearing surface and a thrust bearing surface, and to facilitate the quality control therein.

To achieve the foregoing objects, the present invention is to provide a hydrodynamic type oil-impregnated sintered bearing unit comprising a shaft and a hydrodynamic type oil-impregnated sintered bearing including a bearing body formed of sintered metal, the bearing body being provided with a radial bearing surface opposed to the outer periphery of the shaft via a bearing clearance and being impregnated with oil, the hydrodynamic type oil-impregnated sintered bearing supporting the shaft without contact by means of the hydrodynamic action produced on the radial bearing surface in the relative rotation between the shaft and the bearing body, wherein: at least one bearing end face of the hydrodynamic type oil-impregnated sintered bearing and a flange portion provided on the shaft constitute a thrust bearing section; and the squareness between the aforesaid one bearing end face and the bearing inner periphery and the squareness between the flange portion and the outer periphery of the shaft are controlled to a tolerance that the aforesaid one bearing end face and the flange portion are kept out of uneven contact with each other in the relative rotation between the shaft and the bearing body.

The thrust bearing section of the aforesaid constitution secures surface contact between the rotating side and the stationary side, so that the pressure in the contacting surface can be decreased to prevent wear, thereby avoiding a change in shaft position resulting from the abrasive deformation of the washer as that in a pivot bearing. Besides, the surface contact improves rigidity with respect to moment loads as compared with the point contact in a pivot bearing.

In a thrust bearing section, an insufficient accuracy in a bearing end face or in a flange portion may put the flange portion into not surface contact but uneven contact (point contact or line contact) with the bearing end face. The uneven contact yields a larger torque loss and causes fluctuations in torque, making it impossible to obtain a high rotational accuracy required of information equipment. In this case, even when the bearing end face is provided with a hydrodynamic pressure generating groove so as to keep the thrust bearing section out of contact, the insufficient hydrodynamic effect causes contact and wear between the bearing end face and the flange portion, precluding the improvement in rotational accuracy and durability.

Thus, in the present invention, the squareness between at least one bearing end face and the bearing inner periphery constituting a thrust bearing section and the squareness between the flange portion and the outer periphery of the shaft (in particular, the outer periphery of the shaft opposed to the radial bearing surface) are controlled to a tolerance that the aforesaid one bearing end face and the flange portion are kept out of uneven contact with each other in the relative rotation between the shaft and the bearing body.

Here, in the cases, e.g., where the squareness between the bearing end face and the bearing inner periphery is 4 μm or greater and the squareness between the flange portion and the outer periphery of the shaft is 3 μm or greater, the flange portion may be in not surface contact but uneven contact with the bearing end face. Therefore, the squareness between the bearing end face and the bearing inner periphery is set within 3 μm, and the squareness between the flange portion and the outer periphery of the shaft is set within 2 μm.

In this connection, the "squareness" as employed herein refers to, in the combination of a plane to be a standard and a planar portion to be perpendicular thereto, the magnitude that the planar portion to be perpendicular deviates from a geometrical plane perpendicular to the standard plane.

In a conventional hydrodynamic type oil-impregnated sintered bearing, the bearing end faces thereof are not sufficient in accuracy (the squareness of the bearing end faces relative to the bearing inner periphery is in the order of 10 μm), and it is difficult to mass-produce a bearing body having an accuracy in the aforesaid numerical range. Measures thereto include a method in which, after a bearing is fixed to a housing, the bearing end faces thereof are finished by machining with the bearing inner periphery as the standard, or with the outer periphery or the like being secured in concentricity to the bearing inner periphery as the standard. This, however, gives rise to such problems that: (1) since chips and shavings produced in the machining adhere to the bearing inner periphery, cleaning needs to be performed after the machining; and (2) the additional processes required such as the post machining and the cleaning produce a large increase in cost, thereby harming the reasonability in cost which is the greatest feature of hydrodynamic type oil-impregnated sintered bearings.

Accordingly, in the present invention, a forming die for forming a hydrodynamic pressure generating groove in a radial bearing surface is inserted into the inner periphery of a bearing body material, and a pressing force is applied to the bearing body material while holding both end faces of the bearing body material with a pair of punching surfaces, so that the forming die forms in the inner periphery of the bearing body material a radial bearing surface having the hydrodynamic pressure generating grooves slanting against an axial direction, and at least one of the punching surfaces forms in one end face of the bearing body material a thrust bearing surface constituting a thrust bearing section with a shaft; here, the squareness between the aforesaid at least one punching surface and the outer periphery of the forming die is set within 2 μm (desirably within 1 μm).

As a way to finish the punching surface and the forming die with a high degree of accuracy as mentioned above, the aforesaid one punching surface and the forming die may be constituted integrally. For example, it is possible to adopt such methods that the punch and the forming die are integrally made of an identical member by cutting, or that they are separately fabricated, and integrally fixed by a technique such as pressing-in before finished within 2 μm in squareness between the punching surface and the outer periphery of the forming die. The shaft and the flange portion may be integrally fabricated of an identical member, or they may be separately fabricated before one is pressed into the other, and then finished at a prescribed squareness.

After the bearing body material is formed in a prescribed dimension, the aforesaid forming die is desirably released from the inner periphery of the bearing body material by removing the pressing force to allow spring back of the bearing body material, and producing between the bearing body material and the forming die a difference in thermal expansion such that a difference in dimension widens

between the inner diameter of the bearing body material and the outer diameter of the forming die. This avoids the interference between the forming die and the bearing body material, allowing the forming die to be drawn out of the inner periphery of the bearing body material without breaking the hydrodynamic pressure generating grooves formed.

The aforesaid difference in thermal expansion can be produced by, e.g., applying heat from the bearing body material side after the formation of the bearing surfaces. The forming die typically employs hard metal as its material, which has a coefficient of linear expansion of 5.1×10^{-6} [1/° C.]. Meanwhile, the bearing body material consists mainly of powdered copper and powdered iron, having a coefficient of linear expansion of, for instance, 12.9×10^{-6} [1/° C.]. Accordingly, when the bearing body material is heated to higher temperatures, the difference in dimension between the inner diameter of the bearing body material and the outer diameter of the forming die increases due to the difference in thermal expansion therebetween, facilitating the drawing of the forming die from the bearing body material.

In the aforesaid constitution, it is desirable that the bearing bore diameter d and the bearing length L of the hydrodynamic type oil-impregnated sintered bearing are set as $L\leqq1.2$ d, and the radial bearing surface is provided at one place on the bearing inner periphery. This can achieve thinner models of spindle motors.

In addition, a hydrodynamic pressure generating groove, for feeding oil, slanting against an axial direction may be provided in the bearing inner periphery of the hydrodynamic type oil-impregnated sintered bearing, so that the thrust bearing section is fed with oil by means of the hydrodynamic action produced in the hydrodynamic pressure generating groove. In the case where a bearing end face is a smooth surface having no hydrodynamic pressure generating groove, oil in the thrust bearing section is radially driven by the centrifugal action, possibly resulting in insufficient lubrication especially in high speed rotation and the like. In contrast, the provision of the above-mentioned hydrodynamic pressure generating groove facilitates the oil-film formation in the thrust bearing section to improve the lubricity. In addition, it remarkably lowers the wear in the thrust bearing section, greatly improving the durability.

In this connection, the oil fed to the thrust bearing section is absorbed through the bearing end face and the chambered portions into the inside of the bearing for recovery, and newly fed through the bearing inner periphery to the bearing clearance.

Moreover, in the aforesaid constitution, the thrust bearing section is desirably configured to support the shaft without contact by means of the hydrodynamic action produced in the relative rotation between the shaft and the bearing body. The non-contact support can eliminate the wear in the thrust bearing section to improve the durability yet greatly.

To be concretely, either of the aforesaid one bearing end face and the flange portion opposed thereto constituting the thrust bearing section may be provided with a hydrodynamic pressure generating section having a plurality of concave portions arranged circumferentially (desirably provided at three places or more). In this case, the concave portions serve as oil reservoirs; and when the oil in the concave portions is drawn out to adjacent convex portions with rotation, a pressure is generated, which can increase the film pressure to maintain the thrust bearing section stably in its non-contact state. The concave portions in the hydrodynamic pressure generating section may be hydrodynamic pressure generating grooves having portions slanting against imaginary radial lines drawn on the bearing end face. Here, with rotation, the oil in the thrust bearing section and peripheries thereof is accumulated to the inner periphery side to increase the film pressure, so that the thrust bearing section is maintained more stably in the non-contact state. A spiral-typed or a herringbone-typed shape may be applied to the hydrodynamic pressure generating grooves.

In the aforesaid constitution, it is desirable that the thrust bearing section is arranged at two places axially separated from each other. In this case, thrust loads in both directions can be supported, and the coming-out of the shaft can be prevented as well.

Furthermore, the rate of surface holes of the hydrodynamic type oil-impregnated sintered bearing is desirably set to be 10% or less (desirably 5% or less) in the radial bearing surface, and set to be 5% or less (desirably 2% or less) in the bearing end face constituting the thrust bearing section. The 10%-or-less rate of surface holes in the radial bearing surface can secure the circulation of oil while preventing a pressure drop. At the 5%-or-less rate of surface holes in the aforesaid bearing end face, the running-off of oil through surface holes with arising pressures can be avoided even in the cases where hydrodynamic pressure generating grooves are provided therein. The "surface hole" refers to a portion in which a pore in a porous article's texture opens at the external surface, and the "rate of surface holes" refers to an a real ratio of surface holes by unit area on the external surface.

According to the present invention, it becomes possible for a bearing end face to offer support in the thrust direction. This eliminates the change in shaft position due to a recess in the thrust washer created by the deformation and wear as in a pivot bearing, and makes it possible for even thinner models of the bearing to keep the moment rigidity high. Moreover, in the relative rotation of the shaft and the bearing body, the one bearing end face and the flange portion are out of uneven contact with each other, which yields a smaller torque loss, allowing the suppression of fluctuations in torque to achieve higher rotational accuracies required of information equipment.

In the cases where the thrust bearing section is axially arranged at two places, the axial movement of the rotating shaft can be restrained to improve the impact load characteristics. Particularly, in those cases where a read head and a disc are arranged via a slight gap as in a HDD device, the situation that the head bumps against the disc can be avoided even under an impact load.

Since being constituted of a porous article such as sintered metal alloy, the oil-impregnated sintered bearing can be worked with a high degree of accuracy at a lower cost. Moreover, the bearing being a porous article holds a larger amount of oil, and the circulation of the oil slows its deterioration for improved durability.

To achieve the foregoing objects, the present invention also provides a hydrodynamic type bearing having a radial bearing surface provided in the inner periphery of a bearing body, the radial bearing surface having hydrodynamic pressure generating grooves slanting against an axial direction, the radial bearing surface being opposed via a radial bearing clearance to the outer periphery of a shaft member to be supported, wherein a thrust bearing surface having hydrodynamic pressure generating grooves is formed on at least one end face of the bearing body, simultaneously with the radial bearing surface.

This hydrodynamic type bearing can be constituted by a bearing body formed of sintered metal and impregnated with oil, or a bearing body formed of soft metal.

A hydrodynamic type bearing unit according to the present invention comprises a shaft member having a flange portion, and any of the aforementioned hydrodynamic type bearings, wherein the aforesaid thrust bearing surface and the end face of the flange portion opposed thereto form a thrust bearing clearance.

The aforesaid hydrodynamic type bearing is fabricated in such a manner that a radial bearing surface and a thrust bearing surface each having hydrodynamic pressure generating grooves are simultaneously formed on the inner periphery and at least one end face of a bearing material by: arranging in the inner periphery portion of the bearing material a radial forming die for forming the hydrodynamic pressure generating grooves in the radial bearing surface; holding both end faces of the bearing material with a pair of punching surfaces, at least one of the punching surfaces being provided with a thrust forming die for forming the hydrodynamic pressure generating grooves in the thrust bearing surface; and, in this state, applying a pressing force to the bearing material.

According to the present invention, a hydrodynamic type bearing for non-contact supporting a shaft member in both the radial and thrust directions can be fabricated by a simple method, with a high degree of accuracy and at lower costs. Besides, the simultaneous formation of the radial bearing surface and the thrust bearing surface eliminates the possibility that a bearing surface formed in a preceding process suffers a decrease in accuracy during the following process as in the case of forming the both bearing surfaces in separate processes, and allows the respective bearing surfaces to be formed with a high degree of accuracy at lower costs.

The nature, principle and utility of the invention will become more apparent from the following detailed description when read in conjunction with the accompanying drawings.

BRIEF EXPLANATION OF THE DRAWINGS

In the accompanying drawings:

FIG. 1 is a cross-sectional view of a hydrodynamic type oil-impregnated sintered bearing unit according to the present invention;

FIGS. 2(A) and 2(B) are cross-sectional views showing other embodiments of the present invention;

FIGS. 3(A) and 3(B) are cross-sectional views showing other embodiments of the present invention;

FIG. 4 is a cross-sectional view showing another embodiment of the present invention;

FIG. 5 is a cross-sectional view showing another embodiment of the present invention;

FIGS. 6(A) and 6(B) are views showing another embodiment of the present invention, FIG. 6(A) being a cross-sectional view and FIG. 6(B) being a plan view;

FIGS. 7(A) and 7(B) are views showing another embodiment of the present invention, FIG. 7(A) being a cross-sectional view and FIG. 7(B) being a plan view;

FIGS. 8(A), 8(B) and 8(C) are views showing another embodiment of the present invention, FIG. 8(A) being a cross-sectional view, FIG. 8(B) being a plan view, and FIG. 8(C) being a bottom view;

FIGS. 9(A), 9(B) and 9(C) are views showing another embodiment of the present invention, FIG. 9(A) being a cross-sectional view, and FIGS. 9(B) and 9(C) are bottom views;

FIG. 10 is a side view of a core rod and an upper punch;

FIG. 11 is a sectional view showing the method of the present invention;

FIG. 12 is a cross-sectional view of an optical disc drive incorporating a hydrodynamic type oil-impregnated sintered bearing unit;

FIG. 13 is a cross-sectional view of a conventional hydrodynamic type oil-impregnated sintered bearing unit;

FIG. 14 is a cross-sectional view of a hydrodynamic type oil-impregnated sintered bearing unit;

FIG. 15 is a cross-sectional view of a hydrodynamic type bearing unit according to the present invention;

FIG. 16(A) is a cross-sectional view of a hydrodynamic type bearing according to the present invention, and FIG. 16(B) is a plan view thereof as seen in the direction of b;

FIG. 17 is a schematic cross-sectional view of a molding machine for use in a bearing surface molding process;

FIG. 18 is a front view of a core rod;

FIG. 19 is a cross-sectional view showing the bearing surface molding process;

FIG. 20 is a cross-sectional view showing another embodiment of the hydrodynamic type bearing unit;

FIG. 21 is a cross-sectional view showing another embodiment of the hydrodynamic type bearing unit;

FIG. 22 is a cross-sectional view showing another embodiment of the hydrodynamic type bearing unit; and

FIG. 23 is a cross-sectional view showing another embodiment of the hydrodynamic type bearing unit.

DETAILED DESCRIPTION OF THE PREFERRED EMBODIMENTS

Hereinafter, the preferred embodiments of the present invention will be described with reference to the accompanying drawings.

FIG. 1 is a cross-sectional view of a hydrodynamic type oil-impregnated sintered bearing unit according to the present invention. The bearing unit comprises an oil-impregnated sintered bearing **11** of hydrodynamic type, a cylindrical housing **12** having the oil-impregnated sintered bearing **11** fixed to the bore portion thereof, and a rotating shaft **13** inserted into the bore portion of the oil-impregnated sintered bearing **11**.

The hydrodynamic type oil-impregnated sintered bearing **11** is constituted by impregnating a cylindrical bearing body **11***a* composed of sintered metal with lubricating oil or lubricating grease, the bearing body **11***a* having a radial bearing surface **11***b* opposed to the outer periphery of the rotating shaft **13** via a bearing clearance. The sintered metal bearing body **11***a* is formed of sintered metal consisting mainly of a copper type, an iron type, or the both materials, and desirably employs copper at 20–95% by weight to range from 6.4 to 7.2 g/cm^3 in density. For use as the material of the bearing body **11***a*, cast iron, synthetic resins, ceramics, and the like may be sintered or foam molded into a porous article having a number of pores. The bearing body **11***a* preferably has the rate of surface holes not greater than 10% in its radial bearing surface **11***b*, and that not greater than 5% in the after-mentioned thrust bearing surfaces **11***f***1** and **11***f***2**.

The bearing body **11***a* has only one radial bearing surface **11***b* provided on its inner periphery **11***h*. In the bearing surface **11***b* are circumferentially arranged and formed a plurality of hydrodynamic pressure generating grooves **11***c* (of herringbone type) slanting against the axial direction. As long as they are formed to slant against the axial direction,

the hydrodynamic pressure generating grooves 11*c* may take any shape, e.g. of spiral type, other than that of herringbone type. In the outer periphery of the oil-impregnated sintered bearing 11 is/are formed along the axial direction a groove or a plurality of grooves 11*g*, each serving as an air vent in inserting the shaft 13 into the bore portion of the bearing 11. Incidentally, for thinner models of spindle motors, the bearing bore diameter d and the bearing length L are set to satisfy $L \leqq 1.2$ d.

In the aforesaid oil-impregnated sintered bearing 11, the lubricant (the lubricating oil or the base oil of the lubricating grease) inside the bearing body 11*a* exudes out from the surfaces of the bearing body 11*a* due to the thermal expansion of the oil resulting from a rise in temperature and the generation of pressure caused by the rotation of the rotating shaft 13. The lubricant is then drawn into the bearing clearance by the action of the hydrodynamic pressure generating grooves 11*c*. The oil drawn into the bearing clearance forms a lubricating film to support the rotating shaft without contact. That is, when the aforesaid slanting hydrodynamic pressure generating grooves 11*c* are provided in the bearing surface 11*b*, the hydrodynamic action thereof draws the lubricant having exuded out of the bearing body 11*a* into the bearing clearance, and continuously drives the lubricant into the bearing surface 11*b* as well. This can increase the film pressure to improve the rigidity of the bearing. Incidentally, in assembling the bearing unit, the rotating shaft 13 is desirably inserted into the bearing 11 with the bearing clearance and peripheries of the bearing lubricated so as to be filled with oil.

When a positive pressure is generated in the bearing clearance, the holes existing in the bearing surface 11*b* cause the lubricant to flow back to the inside of the bearing body; however, since new lubricant is successively driven into the bearing clearance, the film pressure and the rigidity are maintained at high levels. The film formed here is continuous and stable, which offers a high rotational accuracy, and lowers shaft run-out, non repeatable run-out(NRRO), jitter, and the like. Besides, the rotating shaft 13 rotates without contact to the bearing body 11*a*, resulting in lower noise and lower costs as well.

The radial bearing surface 11*b* comprises a first groove region m1, a second groove region m2 provided so as to be axially separated from the first region m1, and an annular smooth region n positioned between the two groove regions m1 and m2. In the first groove region m1 are arranged hydrodynamic pressure generating grooves 11*c* slanting in one direction against the axial direction. In the second groove region m2 are arranged hydrodynamic pressure generating grooves 11*c* slanting in the other direction against the axial direction. The hydrodynamic pressure generating grooves 11*c* in the two groove regions m1 and m2 are sectioned by the smooth region n so as to be disconnected from each other. The smooth region n and the portions of ridges 11*e* between hydrodynamic pressure generating grooves 11*c* are at the same level. The hydrodynamic pressure generating grooves 11*c* of such discontinuous type offer the advantages that the accumulation of oil about the smooth region n yields a higher film pressure and that the groove-less smooth region n provides a higher bearing rigidity, as compared with hydrodynamic pressure generating grooves of continuous type, i.e., in which the smooth region n is omitted and hydrodynamic pressure generating grooves 11*c* in the both groove regions m1 and m2 are connected each other into continuous V-shaped grooves.

On one axial end side of the oil-impregnated sintered bearing 11 is provided a thrust bearing section 14. FIG. 1 shows the embodiment in which the thrust bearing section 14 is provided on the upper end side of the oil-impregnated sintered bearing 11. The upper bearing end face 11*f*1 (thrust bearing surface) of the oil-impregnated sintered bearing 11 and a disk-shaped flange portion 13*a* fixed to the rotating shaft 13 are opposed to each other to constitute the thrust bearing section 14. The rotating shaft 13 and the flange portion 13*a* are integrally made from an identical member, or are separately fabricated and then fit to each other. They are finished so that the outer periphery of the rotating shaft 13, especially at the portion opposed to the bearing surface 11*b* when attached to the bearing 11, has a squareness within 2 μm, desirably within 1 μm, with respect to the end face 13*a*1 of the flange portion 13*a* on the bearing 11 side.

Such thrust bearing section 14 offers surface contact to its slide-contacting portions. This avoids fluctuations in shaft position, which come into question in the cases of pivot bearings, and simultaneously enables even the single-row bearing surface 11*b* to increase the moment rigidity to support the shaft with a high degree of accuracy.

Now, in the cases where the flange portion 13*a* is provided on the rotating shaft 13 as described above, it becomes difficult to seal the upper end of the bearing 11 with a conventional seal washer 5 (see FIG. 15) since parts such as a rotor case 8 and a turntable 5 (see FIG. 12) are fixed to the upper end of the rotating shaft 13. On this account, the oil leakage from the upper end of the bearing 11 is capillary sealed by a minute gap between the outer periphery of the flange portion 13*a* and the inner periphery of the housing 12. The sealing gap c is preferably 0.05 mm or less, desirably 0.02 mm or less, and the sealing length a is preferably 0.5 mm or more, desirably 1 mm or more. Oil repellent may be applied to the outer periphery of the flange portion 13*a* or the inner periphery of the housing 12 constituting the seal for more effective prevention of oil leakage.

Meanwhile, the oil leakage from the lower end side of the bearing 11 can be prevented, for example, by pressing a baseplate 15 into the bottom opening portion of the housing 12 and then caulking the same. The gap between the baseplate 15 and the housing 12 may be sealed by adhesive for more effective prevention of oil leakage.

FIG. 2(A) shows an embodiment in which elastic material 15*a* such as resin or rubber is put over the baseplate 15 and used as packing to prevent the oil leakage from the baseplate 15 side. It is also desirable here, if needed, to caulk the baseplate 15 after being pressed into the housing 12.

FIG. 2(B) shows an embodiment in which an annular concave portion 12*a* is provided in the inner periphery of the housing 12 at the area opposed to the outer periphery of the flange portion 13*a*. The rotation of the flange portion 13*a* accumulates oil into the concave portion 12*a* by the centrifugal force, enabling the secure prevention of the oil leakage from the upper end of the housing 12 (centrifugal sealing). Since the centrifugal sealing alone may allow oil leakage in the cases where the shaft position is sideways, the capillary sealing is desirably used in combination.

FIG. 3(A) shows an example in which the outer periphery of the flange portion 13*a* is provided with slanting grooves 13*a*2 that generate air streams toward the bearing 11 side when rotating. The air streams generated drive back oil to the bearing 11 side so that the oil leakage from the upper end of the bearing can be prevented (when not rotating, the capillary sealing prevents the oil leakage). Unlike the hydrodynamic pressure generating grooves 11*c* in the radial bearing surface 11*b*, the above-mentioned slanting grooves 13*a*2 need not be finished with a high degree of accuracy as

long as they can prevent the oil leakage. The grooves have an appropriate depth on the order of 5–30 μm, and can be formed by techniques of rolling and the like.

These slanting grooves **13***a***2**, when formed to run through the width (axial dimension) of the flange portion **13***a* as shown in FIG. 3(A), sometimes feed excessive air into the bearing **11** side; therefore, they may be partially provided as shown in FIG. 3(B). In this case, it is desirable that the annular concave portion **12***a* is provided in the inner periphery of the housing **12** at the area opposed to the region having no slanting grooves **13***a***2** formed.

FIG. 4 shows an embodiment in which the thrust bearing section **14** is constituted by the lower bearing end face **11***f***2** of the bearing **11** and the flange portion **13***a* provided on the shaft end. When rotating, the rotating shaft **13** receives a floating force from the exciting force between the rotor **7***b* and the stator **7***a* (see FIG. **12**) to float over the baseplate **15**, and the thrust force is supported by the lower bearing end face **11***f***2** (thrust bearing surface) and the upper surface **13***b***2** of the flange portion **13***b*. In the upper surface of the baseplate **15** and immediately below the rotating shaft **13** is arranged a thrust washer **15***a* consisting of resin material or the like having high lubricity, so as to reduce the friction against the shaft end immediately after the start of and immediately before the stop of the motor. The top opening of the housing **12** is blocked with a seal washer **16** for preventing oil leakage, and the gap between the washer and the shaft is provided to be 0.2 mm or less to prevent the outward leakage of oil (capillary sealing). Oil repellent may be applied to the inner periphery of the seal washer **16**, the upper and lower surfaces of the inner peripheral portion thereof, or the outer periphery of the shaft **13** facing the inner periphery of the seal washer **16**, for more effective prevention of oil leakage.

FIG. **5** shows an embodiment in which the bearing inner periphery **11***h* of the hydrodynamic type oil-impregnated sintered bearing **11** is provided with hydrodynamic pressure generating grooves **11***j* for oil supply slanting against the axial direction, so as to supply oil to the thrust bearing section **14** by means of the hydrodynamic action created by the hydrodynamic pressure generating grooves **11***j*. The hydrodynamic pressure generating grooves **11***j* are connected with the hydrodynamic pressure generating grooves **11***c* in the radial bearing surface **11***b* at the groove region (on the thrust bearing section **14** side), being formed into V-shapes. The provision of the hydrodynamic pressure generating grooves **11***j* for oil supply facilitates the formation of the oil film in the thrust bearing section **14** to improve the lubricity; in addition, it remarkably reduces wear in the thrust bearing section **14**, greatly improving the durability.

FIGS. 6(A) through 7(B) show embodiments in which the thrust bearing section **14** is provided to non-contact support the rotating shaft **13** by means of the hydrodynamic action generated when rotating the rotating shaft **13**. The non-contact support eliminates the friction in the thrust bearing section **14** to greatly improve the durability. The hydrodynamic action can be obtained by the provision of a hydrodynamic pressure generating section **17**, which has a plurality of concave portions **11***k* arranged circumferentially in either of the bearing end face **11***f***1** and the flange portion **13***a* opposed thereto constituting the thrust bearing section **14**. The concave portions **11***k* include, for example, hydrodynamic pressure generating grooves.

FIGS. 6(A) and 6(B) show an example of the thrust bearing section **14** having the hydrodynamic pressure generating section **17**, in which the thrust bearing surface **11***f***1** is provided with hydrodynamic pressure generating grooves **11***k* having portions slanting against imaginary radial lines drawn on the bearing end face. The hydrodynamic pressure generating grooves **11***k* are of herringbone type, that is, of V-shapes having the curved sections generally at the radial intermediate portions thereof. The hydrodynamic pressure generating grooves are arranged and formed circumferentially at equal intervals. In this case, with rotation, the oil in the thrust bearing section **14** and peripheral portions thereof is accumulated about the curved sections to increase the film pressure, so that the thrust bearing section **14** can be stably maintained in its non-contact state. In addition to the herringbone type, the spiral type is also applicable to the shape of the hydrodynamic pressure generating grooves. Moreover, the hydrodynamic pressure generating grooves **11***k* may be provided in the end face **13***a***1** of the flange portion to form the thrust bearing surface **11***f***1** as a smooth surface having no hydrodynamic pressure generating groove.

FIGS. **7**(A) and **7**(B) show an embodiment in which the hydrodynamic pressure generating grooves **11***k* are provided in the thrust bearing surface **11***f***1** as in FIGS. 6(A) and 6(B), and the hydrodynamic pressure generating grooves **11***j* for oil supply are provided in the bearing inner periphery as in FIG. **5**.

FIGS. **8**(A) through **9**(C) show embodiments in which thrust bearing sections **14***a* and **14***b* are provided at two places axially separated from each other, so as to support thrust loads in both directions. In FIGS. 8(A) to 8(C), flange portions **13***a* and **13***b* are arranged on both end sides of the bearing, so that the two thrust bearing sections **14***a* and **14***b* formed between the flange portions **13***a*, **13***b* and the both bearing end face **11***f***1**, **11***f***2** can support the thrust loads in the both directions. As shown in FIGS. 8(B) and 8(C), in either the thrust bearing surfaces **11***f***1**, **11***f***2** or end surfaces of the flange portions **13***a*, **13***b* opposed thereto both constituting the thrust bearing sections (in these figures, in the thrust bearing surfaces **11***f***1**, **11***f***2**) are formed the same hydrodynamic pressure generating grooves **11***k* as those in FIGS. 6(A) and 6(B). This configuration can not only offer the thrust support in the both directions, but also prevent the rotating shaft **13** from coming out of the bearing **11**; therefore, damage to the motor can be avoided even when an impact load is imposed on the rotating shaft **13**.

FIGS. 9(A) to 9(C) show an embodiment in which a flange **13***b* is provided between the bearing **11** and the baseplate **15**, and the thrust bearing sections **14***a* and **14***b* are constituted on both sides of the flange portion **13***b*. That is, in either the upper end face **13***b***1** of the flange portion **13***b* or the lower bearing end face **11***f***2** and in either the lower end face **13***b***2** of the flange portion **13***b* or the upper surface of the baseplate **15** (in these figures, in the lower bearing end face **11***f***2** and in the lower end face **13***b***2** of the flange portion) are arranged the same hydrodynamic pressure generating grooves **11***k* as those in FIGS. 6(A) and 6(B), offering the same effect as that of the configuration in FIGS. 8(A) to 8(C).

The bearing body **11***a* of the aforesaid hydrodynamic type oil-impregnated sintered bearing **11** can be fabricated by applying, e.g., sizing, rotational sizing, and bearing surface molding to cylindrical sintered metal material (bearing body material) obtained by compression molding the aforesaid powdered metal and sintering the same.

The sizing process is a process for sizing the outer periphery and the inner periphery of the sintered metal material to correct the bend and the like generated in the

sintering process, and is performed by pressing the outer periphery of the sintered metal material into a cylindrical die while pressing a sizing pin into the inner periphery of the material. The rotational sizing process is a process in which a rotational sizing pin of generally polygonal section (obtained by partially leveling the outer periphery of a circular-section pin, leaving arc portions at circumferential symmetric positions) is pressed against the inner periphery of the sintered metal material while the sizing pin is rotated to perform the sizing of the inner periphery. This rotational sizing corrects the inner periphery of the sintered metal material in roundness and cylindricity, and finishes the same at the rate of surface holes of, for example, 3–15%. The bearing surface molding process is a process in which a forming die having the shape corresponding to the bearing surface of a finished product is pressed against the inner periphery of the sintered metal material having had the sizing processes applied as described above, so as to simultaneously mold the forming region of the hydrodynamic pressure generating grooves and the other regions (the ridges **11***e* and the annular smooth region n) on the bearing surface.

FIG. **11** illustrates by example the general configuration of a molding machine for use in the bearing surface molding process. This machine is composed mainly of: a cylindrical die **20** into which the outer periphery of sintered metal material **11'** is pressed; a core rod **21** of hard metal for molding the inner periphery of the sintered metal material **11'**; and upper and lower punches **22** and **23** for pressing both end faces of the sintered metal material **11'** from above and below. The core rod **21** and the upper punch **22** are integrated, and the outer periphery of the core rod **21** and the punching surface of the upper punch **22** are finished within 2 μm in squareness.

As shown in FIG. **10**, the outer periphery of the core rod **21** is provided with a forming die **21***a* having the concave and convex portions corresponding to the bearing surface **11***b* of a finished product in shape. The convex portion **21***a***1** of the forming die **21***a* is to form the regions of the hydrodynamic pressure generating grooves **11***c* in the bearing surface **11***b*, and the concave portion **21***a***2** is to form the regions other than the hydrodynamic pressure generating grooves **11***c* (the ridges **11**e and the annular smooth region n). The difference in dimension between the convex portions **21***a***1** and the concave portions **21***a***2** in the forming die **21***a* is as nearly equally minute (for example, on the order of 2–5 μm) as the depth of the hydrodynamic pressure generating grooves **11***c*, while it is considerably exaggerated in the figure. In this connection, in the cases of providing the hydrodynamic pressure generating grooves **11***k* in the upper and lower bearing end faces **11***f***1**, **11***f***2** (see FIGS. 6(A) through 9(C)), the punching surfaces **22***a*, **23***a* of the upper and lower punches **22**, **23** are also provided with forming dies for transfer having the shapes corresponding to the aforesaid hydrodynamic pressure generating grooves **11***k*.

This molding machine performs the molding in accordance with the procedures ① to ④ shown in FIG. **11**.

For a start, the sintered metal material **11'** is positioned and placed on the upper surface of the die **20**. Subsequently, the upper punch **22** and the core rod **21** are lowered, so that the sintered metal material **11'** is pressed into the die **20** and then pressed against the lower punch **23** for compression from above and below (①).

The sintered metal material **11'** is deformed under the pressing forces from the die **20** and the upper and lower punches **22**, **23**, and the inner periphery thereof is pressed to the forming die **21***a* on the core rod **21**. This transfers the shape of the forming die **21***a* to the inner periphery of the sintered metal material **11'**, molding the bearing surface **11***b* in a prescribed shape and dimension (at the same time, the outer periphery and the both end faces of the sintered metal material **11'** are sized as well).

After the molding of the bearing surface **11***b* is completed, the upper and lower punches **22**, **23** and the core rod **21** are simultaneously lifted while holding the physical relationship between the sintered metal material **11'** and the core rod **21** (②), drawing the sintered metal material **11'** out of the die **20**. Subsequently, the outer periphery of the sintered metal material **11'** being clamped by a clamper **24** is subjected to heated air from a heater **25** such as a heated air generator to heat the sintered metal material **11'** (③), and then the sintered metal material **11'** is released from the core rod **21** (④). Here, as soon as the sintered metal material **11'** is drawn out of the die **20**, the sintered metal material **11'** yields spring back to expand in dimension of its inner diameter. Besides, the sintered metal material is elevated higher in temperature than the core rod **21** by the heating, and the sintered metal material **11'** (consisting mainly of copper) is greater in coefficient of thermal expansion than the core rod **21** (made of hard metal alloy); therefore, the sintered metal material **11'** further expands in the dimension of the inner diameter. Thus, the interference between the core rod **21** and the sintered metal material **11'** is avoided, allowing the core rod **21** to be drawn out of the inner periphery of the sintered metal material **11'** without breaking the hydrodynamic pressure generating grooves **11***c*. The heating process by the heater **25** may be omitted in the cases where the sintered metal material **11'** is smoothly releasable merely by means of the spring back.

The sintered metal material **11'** fabricated through the above-described processes is subjected to cleaning, and impregnated with lubricating oil or lubricating grease so as to hold oil, completing the hydrodynamic type sliding bearing (hydrodynamic type oil-impregnated porous bearing) shown in FIG. **1**. This bearing **11** is fixed to the inner periphery of the housing **12** by e.g., adhesion. Incidentally, after the attachment of the bearing **11** to the housing **12**, the bearing clearance and spaces around the bearing can be filled with oil, in addition to the impregnated oil, to greatly improve the lubricity.

When the squareness between the outer periphery of the core rod **21** and the punching surface **22***a* of the upper punch **22** is set within 2 μm as described above, it becomes possible to provide the oil-impregnated sintered bearing **11** within 3 μm in squareness of the thrust bearing surface **11***f***1** with respect to the bearing inner periphery **11***h*. This bearing **11** and a rotating shaft **13** having the squareness between the flange portion **13***a* and its outer periphery set in a prescribed range can be combined to avoid uneven contact in the thrust bearing section **14**, achieving secure surface contact.

FIG. **15** is a cross-sectional view of a hydrodynamic type bearing unit according to the present invention. This bearing unit comprises a hydrodynamic type bearing **1**, a generally cylindrical housing **2** having the hydrodynamic type bearing **1** fixed to the bore portion thereof, and a shaft member **3** inserted into the bore portion of the hydrodynamic type bearing **1**. On one end of the shaft member **3**, an axially-projecting flange portion **3***a* is provided by a method of integral molding, pressing-in of another member, or the like. This flange portion **3***a* is arranged so as to be accommodated between a baseplate **4** sealing one opening of the housing **2** and one end face of the hydrodynamic type bearing **1**. The other opening of the housing **12** is blocked with a sealing member **5** such as a seal washer to prevent outward leakage of oil.

The hydrodynamic type bearing 1 of this embodiment is a hydrodynamic type oil-impregnated sintered bearing in which its bearing body 10 composed of cylindrical sintered metal is impregnated with lubricating oil or lubricating grease. The bearing body 10 is formed of sintered metal consisting mainly of a copper type, an iron type, or the both materials, and desirably molded by using copper at 20–95% by weight.

The inner periphery of the bearing body is provided with a radial bearing surface 10*r*, which radially supports without contact the shaft member 3 functioning as a rotating shaft. The radial bearing surface 10*r* is opposed to the outer periphery of the shaft member 3 via a radial bearing clearance Cr; the present embodiment illustrates a case in which a pair of radial bearing surfaces 10*r* are provided so as to be axially separated, as shown in FIGS. 16(A) and 16(B). In both the radial bearing surfaces 10*r* are circumferentially arranged and formed a plurality of hydrodynamic pressure generating grooves 11 (of herringbone type) slanting against the axial direction. As long as being formed to slant against the axial direction, the hydrodynamic pressure generating grooves 11 may take any shape, e.g. of spiral type, other than that of herringbone type. In the outer periphery of the oil-impregnated sintered bearing 1 is/are formed a groove or a plurality (two, in the figures) of grooves 12 along the axial direction. The grooves 12 function as air vents for securing air communication from the space enclosed with the bearing body 10 and the baseplate 4 to the exterior thereof when the hydrodynamic type bearing 1 is attached to the housing 2 as shown in FIG. 15.

The radial bearing surfaces 10*r* each comprises a first groove region m1, a second groove region m2 provided so as to be axially separated from the first region m1, and an annular smooth region n positioned between the two groove regions m1 and m2. In the first groove region m1 are arranged hydrodynamic pressure generating grooves 11 slanting in one direction against the axial direction. In the second groove region m2 are arranged hydrodynamic pressure generating grooves 11 slanting in the other direction against the axial direction. The hydrodynamic pressure generating grooves 11 in the two groove regions m1 and m2 are sectioned by the smooth region n so as to be disconnected from each other. The smooth regions n and the portions of ridges 13 between hydrodynamic pressure generating grooves 11 are at the same level. The hydrodynamic pressure generating grooves 11 of such discontinuous type offer the advantages that the accumulation of oil about the smooth regions n yields a higher film pressure and that the groove-less smooth regions n provide a higher bearing rigidity, as compared with hydrodynamic pressure generating grooves of continuous type, i.e., in which the smooth regions n are omitted and hydrodynamic pressure generating grooves 11 in the both groove regions m1 and m2 are connected each other into continuous V-shaped grooves.

In the aforesaid oil-impregnated sintered bearing 1, the lubricant (the lubricating oil or the base oil of the lubricating grease) inside the bearing body 10 exudes out from the surfaces of the bearing body 10 due to the thermal expansion of the oil resulting from the generation of pressure and a rise in temperature with the rotation of the shaft member 3. The lubricant is then drawn into the radial bearing clearance Cr by the action of the hydrodynamic pressure generating grooves 11. The oil drawn into the radial bearing clearance Cr forms a lubricating film to support the shaft member 3 without contact. That is, the hydrodynamic action of the aforesaid slanting hydrodynamic pressure generating grooves 11 draws the lubricant having exuded out of the bearing body 10 into the radial bearing clearance Cr, and continuously drives the lubricant into the radial bearing surfaces 10*r* as well. This can increase the film pressure to improve the rigidity of the bearing. When a positive pressure is generated in the radial bearing clearance Cr, the holes existing in the radial bearing surfaces 10*r* cause the lubricant to flow back to the inside of the bearing body 10; however, since new lubricant is successively driven into the radial bearing clearances 10*r*, the film pressure and the rigidity are maintained at high levels. The film formed here is continuous and stable, which offers a high rotational accuracy, and lowers shaft run-out, non repetitive readout overall (NRRO), jitter, and the like. Besides, the shaft member 3 rotates without contact to the bearing body 11*a*, resulting in lower noise and lower costs as well.

One end face of the bearing body 10 (the end face opposed to the flange portion 3*a* of the shaft member 3) is provided with a thrust bearing surface 10*s*, which has been molded simultaneously with the radial bearing surfaces 10*r*. In the thrust bearing surface 10*s*, a plurality of hydrodynamic pressure generating grooves 14 having portions slating against imaginary radial lines drawn on the bearing end face are arranged and formed circumferentially at equal intervals. In the present embodiment illustrates, those of herringbone type, i.e., of approximate V-shapes having the curved sections generally at the radial intermediate portions thereof are illustrated as an example of the hydrodynamic pressure generating grooves 14; however, any other shape can be applied thereto as long as meeting the above-mentioned conditions.

In the bearing unit shown in FIG. 15, the shaft member 3, when rotating, receives a floating force from the exciting force between the rotor 8 and the stator 7 (see FIG. 23) to float over the baseplate 4. Here, by the same action as described above, a hydrodynamic oil film is formed in a thrust bearing clearance Cs between the thrust bearing surface 10*s* and the end face of the flange portion 3*a* opposed thereto, so that the shaft member 3 is non-contact supported in the thrust direction. In the upper surface of the baseplate 4 and immediately below the shaft member 3 is arranged a thrust washer 4*a* consisting of resin material or the like having high lubricity, so as to reduce the friction against the shaft end immediately after the start of and immediately before the stop of the motor.

The bearing body 10 of the aforesaid hydrodynamic type oil-impregnated sintered bearing 1 can be fabricated by applying, e.g., sizing, rotational sizing, and bearing surface molding to cylindrical sintered metal material (bearing material) obtained by compression molding the aforesaid powdered metal and sintering the same.

The sizing process is a process for sizing the outer periphery and the inner periphery of the sintered metal material to correct the bend and the like generated in the sintering process, and is performed by pressing the outer periphery of the sintered metal material into a cylindrical die while pressing a sizing pin into the inner periphery of the material. The rotational sizing process is a process in which a rotational sizing pin of generally polygonal section (obtained by partially leveling the outer periphery of a circular-section pin, leaving arc portions at circumferential symmetric positions) is pressed against the inner periphery of the sintered metal material while the sizing pin is rotated to perform the sizing of the inner periphery. This rotational sizing corrects the inner periphery of the sintered metal material in roundness and cylindricity, and finishes the same at the rate of surface holes of, for example, 3–15%. The bearing surface molding process is a process in which

forming dies having the shapes corresponding to the radial bearing surfaces **10r** and the thrust bearing surface **10s** are pressed against the inner periphery and at least one end face of the sintered metal material having had the sizing processes applied as described above, so as to simultaneously mold the regions of the hydrodynamic pressure generating grooves **14** and the other regions (for example, the ridges **13** and the annular smooth regions n in the radial bearing surfaces **10r**) on the bearing surfaces **10r** and **10s**.

FIG. **17** illustrates by example the general configuration of a molding machine for use in the bearing surface molding process. This machine is composed mainly of a cylindrical die **20** for molding the outer periphery of sintered metal material **10'**, a core rod **21** of hard metal for molding the inner periphery of the sintered metal material **10'**, and upper and lower punches **22** and **23** for pressing both end faces of the sintered metal material **10'** from above and below.

As shown in FIG. **18**, the outer periphery of the core rod **21** is provided with a forming die **21a** (radial forming die) having the concave and convex portions corresponding to the pair of radial bearing surfaces **10r** in shape. The convex portion **21a1** of the forming die **21a** is to form the regions of the hydrodynamic pressure generating grooves **11** in the radial bearing surfaces **10r**, and the concave portion **21a2** is to form the regions other than the hydrodynamic pressure generating grooves **11** (the ridges **13** and the annular smooth regions n). The difference in dimension between the convex portions **21a1** and the concave portions **21a2** in the forming die **21a** is as nearly equally minute (for example, on the order of 2–5 μm) as the depth of the hydrodynamic pressure generating grooves **11** in the radial bearing surfaces **10r**, while it is considerably exaggerated in the figure. Moreover, the punching surface of either one punch (for example, the upper punch **22**) is provided with a forming die **22a** (thrust forming die) having the concave and convex portions corresponding to the hydrodynamic pressure generating grooves **14** in the thrust bearing surface **10s**. Which of the upper and lower punches **22, 23** the thrust forming die is provided on can be freely decided in accordance with such factors as the handleability of the work in subsequent processes, and a thrust forming die **23a** may be provided in the lower punch **23** in contradiction to the above-described case. While the concrete shape of the thrust forming die **22a** (or the thrust forming die **23a**) is not illustrated, it will be understood that the thrust forming die forms the regions of the hydrodynamic pressure generating grooves **14** in the thrust bearing surface **10s** with its convex portions, and forms the regions other than the hydrodynamic pressure generating grooves **14** with its concave portions, like the radial forming die **21a**. This molding machine performs the molding in accordance with the procedures ① to ④ shown in FIG. **19**.

For a start, the sintered metal material **10'** is positioned and placed on the upper surface of the die **20**. Subsequently, the upper punch **22** and the core rod **21** are lowered, so that the sintered metal material **10'** is pressed into the die **20** and then pressed against the lower punch **23** for compression from above and below (①).

The sintered metal material **10'** is deformed under the pressing forces from the die **20** and the upper and lower punches **22, 23**, and the inner periphery and the one end face are pressed to the forming die **21a** on the core rod **21** and the forming die **22a** on the upper punch **22**, respectively. This transfers the shapes of the forming dies **21a** and **22a** to the inner periphery and the one end face of the sintered metal material **10'**, simultaneously molding the radial bearing surfaces **10r** and the thrust bearing surface **10s** in prescribed shapes and dimensions (at the same time, the outer periphery and the both end faces of the sintered metal material **10'** are sized as well).

After the molding of the both bearing surfaces **10r** and **10s** is completed, the upper and lower punches **22, 23** and the core rod **21** are lifted integrally while holding the physical relationship between the sintered metal material **10'** and the core rod **21** (②), drawing the sintered metal material **10'** out of the die **20**. Subsequently, the outer periphery of the sintered metal material **10'** is subjected to heated air from a heater such as a heated air generator to heat the sintered metal material **10'** (③), and then the sintered metal material **10'** is released from the core rod **21** (④). Here, as soon as the sintered metal material **10'** is drawn out of the die **20**, the sintered metal material **10'** yields springback to expand in dimension of the inner diameter. Moreover, since the sintered metal material **10'** is elevated higher in temperature than the core rod **21** by the heating and the sintered metal material **10'** (consisting mainly of copper) is greater in coefficient of thermal expansion than the core rod **21** (made of hard metal alloy), the sintered metal material **10'** further expands in the dimension of the inner diameter. Thus, the interference between the core rod **21** and the sintered metal material **10'** is avoided, allowing the core rod **21** to be drawn out of the inner periphery of the sintered metal material **10'** without breaking the hydrodynamic pressure generating grooves **11**. The heating process by the heater may be omitted in the cases where the sintered metal material **10'** is smoothly releasable merely by means of the springback.

The sintered metal material **10'** fabricated through the above-described processes is subjected to cleaning, and impregnated with lubricating oil or lubricating grease so as to hold oil, completing the oil-impregnated sintered bearing **1** shown in FIGS. **16(A)** and **16(B)**. This bearing **1** is fixed to the inner periphery of the housing **2** by e.g., adhesion. Incidentally, after the attachment of the bearing **1** to the housing **2**, the respective bearing clearances Cr, Cs and spaces around the bearing can be filled with oil, in addition to the impregnated oil, to greatly improve the lubricity.

The forming of the bearing surfaces **10r, 10s** by compression molding the sintered metal material **10'** as described above can simplify the processes, lowering the production costs through the shortening of cycle time and the improvement in mass productivity. Moreover, merely performing the final process of bearing surface molding (hydrodynamic sizing) can readily produce hydrodynamic type bearings with a high degree of accuracy, facilitating the quality control. It is also easy to simultaneously mold the radial bearing surfaces **10r** and the thrust bearing surface **10s**; in this case, the problems can be avoided which rise in the cases of molding the bearing surfaces **10r, 10s** in separate processes, that is, the problems of accuracy decrease in bearing surfaces molded in a preceding process, and the like.

FIGS. **20** through **23** show other embodiments employing the aforesaid hydrodynamic type oil-impregnated sintered bearing **1**.

FIG. **20** shows an embodiment in which the other end face of the bearing body **10** (on the opening side of the housing **2**) is provided with the above-mentioned thrust bearing surface **10s**. Here, the thrust bearing clearance Cs is formed between the aforesaid thrust bearing surface **10s** and the end face (lower end face) of the flange portion **3a** arranged on the shaft member **3**. In the figure, elastic material **4b** such as resin or rubber is put over the baseplate **4** and used as packing to prevent the oil leakage through the joining portion between the baseplate **4** and the housing **2**.

FIG. 21 shows an embodiment in which both end faces of the bearing body **10** are provided with thrust bearing surfaces **10s1** and **10s2**, respectively. The both end faces of the bearing body **10** are opposed to end faces of two flange portions *3a1*, *3a2*, which are provided at two places on the shaft member **3**, via thrust bearing clearances Cs1, Cs2, respectively. In this case, since the support of both-directional thrust loads is made possible and the shaft member **3** is prevented from coming out, damage to the motor can be avoided when an impact load is imposed on the shaft member **3**. Thrust forming dies *22a*, *23a* having the concave and convex portions corresponding to the shapes of the hydrodynamic pressure generating grooves can be provided on the punching surfaces of the upper and lower punches **22, 23** in FIG. **17** so that the thrust bearing surfaces **10s1** and **10s2** are molded simultaneously with the radial bearing surfaces **10r** by exactly the same procedures as those in FIG. **19**.

FIG. 22 shows an embodiment in which the aforesaid thrust bearing surface **10s1** is provided on one end face of the bearing body **10** (on the bottom side of the housing **2**) as in FIG. **15**, and such thrust bearing surface **10s2** is provided on either the opposing surfaces of the flange portion *3a* or the baseplate **4** (for example, on the upper surface of the baseplate **4**), offering the same effect as that of the configuration in FIG. **21**.

FIG. 23 shows an embodiment in which the housing **2** and the baseplate **4** shown in FIG. **22** are integrated into a closed-bottomed cylindrical housing **2'** (bag-shaped housing). On one end face of the bearing body **10** and on either of the opposing surfaces of the flange portion *3a* and a housing bottom surface *2a* (for example, the housing bottom surface *2a*) are provide the aforesaid thrust bearing surfaces **10s1** and **10s2** (here, the radial bearing clearance Cr and the thrust bearing clearances Cs1, Cs2 are shown exaggerated in width). Here, in addition to the same effect as that of the configuration in FIG. **21**, further cost-lowering and the like can be achieved by the complete prevention of the oil leakage through the joining portion between the baseplate **4** and the housing **2**, and by the reduction of the number of component parts. Such bag-shaped housing **2'** can be applied to the bearing units in FIGS. **15, 20,** and **21** to offer the same effect. In the figure, the reference numeral **6** designates a disc hub holding an optical disc or the like and being connected to the top end of the shaft member **3**, the reference numeral **7** a motor stator fixed to the bag-shaped housing **2'**, and the reference numeral **8** a motor rotor fixed to the disc hub **6**.

While the above description has illustrated by examples the cases in which the bearing body **10** is formed of sintered metal, the present invention is also applicable in the case where the bearing body **10** is formed of soft metals such as aluminum, brass, and bronze. Here, the radial bearing surfaces **10r** and the thrust bearing surface **10s** can be simultaneously molded by the same procedures as those shown in FIGS. **17** through **19**. In a case where the bearing material, after the bearing surface molding, is hard to release from the core rod **21**, the bearing material should be heated at the process ③. Here, lubricating oil is filled into the radial bearing clearance Cr and the thrust bearing clearance Cs as the lubricant.

While there has been described what are at present considered to be preferred embodiments of the invention, it will be understood that various modifications may be made thereto, and it is intended that the appended claims cover all such modifications as fall within the true spirit and scope of the invention.

What is claimed is:

1. A hydrodynamic type oil-impregnated sintered bearing unit comprising a shaft and a hydrodynamic type oil-impregnated sintered bearing including a bearing body formed of sintered metal, said bearing body being provided with a radial bearing surface opposed to an outer periphery of said shaft via a bearing clearance and being impregnated with oil, said hydrodynamic type oil-impregnated sintered bearing supporting said shaft without contact by means of hydrodynamic action produced on said radial bearing surface in a relative rotation between said shaft and said bearing body, wherein:

at least one bearing end face of said hydrodynamic type oil-impregnated sintered bearing and a flange portion provided on said shaft constitute a thrust bearing section; and

a squareness between said one bearing end face and a bearing inner periphery and a squareness between said flange portion and the outer periphery of said shaft are controlled to a tolerance that said one bearing end face and said flange portion are kept out of uneven contact with each other in a relative rotation between said shaft and said bearing body.

2. The hydrodynamic type oil-impregnated sintered bearing unit according to claim **1**, wherein:

the squareness between said one bearing end face and the bearing inner periphery is set within 3 μm; and

the squareness between said flange portion and the outer periphery of said shaft is set within 2 μm.

3. The hydrodynamic type oil-impregnated sintered bearing unit according to claim **1** or **2**, wherein:

a bearing bore diameter d and a bearing length L of said hydrodynamic type oil-impregnated sintered bearing are set as

$L \leqq 1.2d$; and

said radial bearing surface is provided at one place on the bearing inner periphery.

4. The hydrodynamic type oil-impregnated sintered bearing unit according to claim **1** or **2**, wherein:

a hydrodynamic pressure generating groove, for feeding oil, slanting against an axial direction is provided in the bearing inner periphery of said hydrodynamic type oil-impregnated sintered bearing, so that said thrust bearing section is fed with oil by means of hydrodynamic action produced in said hydrodynamic pressure generating grooves.

5. The hydrodynamic type oil-impregnated sintered bearing unit according to claim **1** or **2**, wherein

said thrust bearing section supports said shaft without contact by means of hydrodynamic action produced in the relative rotation between said shaft and said bearing body.

6. The hydrodynamic type oil-impregnated sintered bearing unit according to claim **5**, wherein

either of said one bearing end face and said flange portion opposed thereto constituting said thrust bearing section is provided with a hydrodynamic pressure generating section having a plurality of concave portions arranged circumferentially.

7. The hydrodynamic type oil-impregnated sintered bearing unit according to claim **6**, wherein

said concave portions in said hydrodynamic pressure generating section are hydrodynamic pressure generating grooves having portions slanting against imaginary radial lines drawn on said bearing end face.

8. The hydrodynamic type oil-impregnated sintered bearing unit according to claim **1** or **2**, wherein

said thrust bearing section is axially arranged at two places to support thrust loads in both directions.

9. The hydrodynamic type oil-impregnated sintered bearing unit according to claim **1** or **2**, wherein

a rate of surface holes of said hydrodynamic type oil-impregnated sintered bearing is set to be 10% or less in said radial bearing surface, and set to be 5% or less in said bearing end face constituting said thrust bearing section.

10. A hydrodynamic type bearing having a radial bearing surface provided in an inner periphery of a bearing body, said radial bearing surface having a hydrodynamic pressure generating groove slanting against an axial direction, said radial bearing surface being opposed via a radial bearing clearance to an outer periphery of a shaft member to be supported, wherein

a thrust bearing surface having a hydrodynamic pressure generating groove is formed on at least one end face of said bearing body, simultaneously with said radial bearing surface, wherein both of said radial bearing surface and said thrust bearing surface are pressed surfaces formed bv a forming die having a shape corresponding to the bearing surface of a finished product.

11. The hydrodynamic type bearing according to claim **10**, wherein

said bearing body is formed of sintered metal and impregnated with oil.

12. The hydrodynamic type bearing according to claim **10**, wherein

said bearing body is formed of soft metal.

13. A hydrodynamic type bearing unit comprising a shaft member having a flange portion, and a hydrodynamic type bearing according to any one of claims **10** through **12**, wherein

said thrust bearing surface and the end face of said flange portion opposed thereto form a thrust bearing clearance.

* * * * *

US006086257A

United States Patent [19]

Lee

[11] Patent Number: 6,086,257

[45] Date of Patent: Jul. 11, 2000

[54] **SLIDING BEARING AND MANUFACTURING METHOD THEREOF**

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[21] Appl. No.: **09/061,927**

[22] Filed: **Apr. 17, 1998**

[30] **Foreign Application Priority Data**

Apr. 19, 1997	[KR]	Rep. of Korea	97-14658
Jul. 16, 1997	[KR]	Rep. of Korea	97-33090
Mar. 6, 1998	[KR]	Rep. of Korea	98-7387

[51] **Int. Cl.**^{7} **F16C 17/00**; F16C 33/02; B22F 9/00

[52] **U.S. Cl.** **384/279**; 384/910; 384/129; 75/246

[58] **Field of Search** 384/129, 276, 384/280, 286, 291, 289, 322, 279, 397, 902, 910, 912, 913; 75/246, 231; 428/553; 252/34; 308/240

[56] **References Cited**

U.S. PATENT DOCUMENTS

2,086,787	7/1937	Whiteley	384/370
3,945,695	3/1976	Speakman	308/240
4,233,071	11/1980	Bierlein et al.	75/231
5,254,273	10/1993	Kageyama et al.	252/34

Primary Examiner—David A. Bucci
Assistant Examiner—Colby Hansen
Attorney, Agent, or Firm—Knobbe, Martens, Olson & Bear LLP

[57] **ABSTRACT**

A sliding bearing having an improved load-carrying capacity and fatigue strength, by sintering a molded body of iron-based metal mixture powder and concurrently bonding the same to steel-backed metal layer, in which a lubricating oil having the optimal lubrication conditions is impregnated, and providing convenient maintenance and management thereof due to an increased grease non-supplying period, and a manufacturing method of the same. A sliding bearing includes a steel-backed metal layer, and an iron-based sintered alloy layer formed of copper of 10–30 wt % and iron for the residue. The iron-based sintered alloy layer is sintered and concurrently bonded to the steel-backed metal layer. A bush type sliding bearing is formed by inserting a steel-backed metal layer into a mold of a press having cavity for installation of a core at the central portion of the mold and a lower pressing member at the lower portion thereof; filling metal mixture powder formed of copper of 10–30 wt %, graphite of 6.5 wt % or less, and iron for the residue between the core and the steel-backed metal layer and then inserting an upper pressing member into the upper portion of the mold; forming the metal mixture powder into a molded body by selectively applying pressure of 50–300 kgf/cm^2 to the upper pressing member and/or the lower pressing member, respectively; and sintering the molded body into a sintered alloy layer by maintaining the molded body together with the steel-backed metal layer under predetermined gas atmosphere at temperature of 1,065° C.–1,095° C. for 3–25 minutes and concurrently bonding the sintered alloy layer to the steel-backed metal layer.

8 Claims, 10 Drawing Sheets

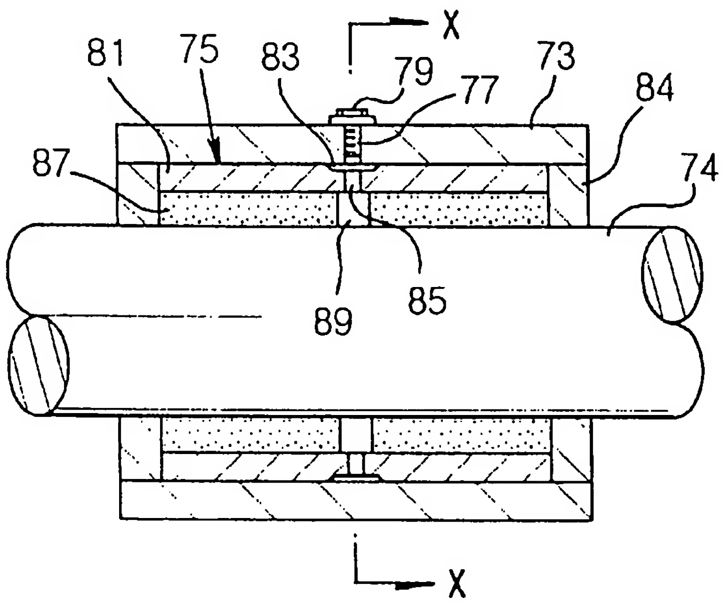


Fig. 1

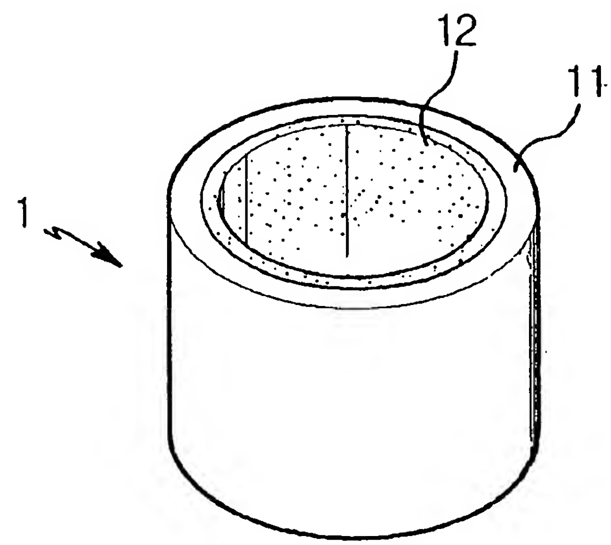


Fig. 2

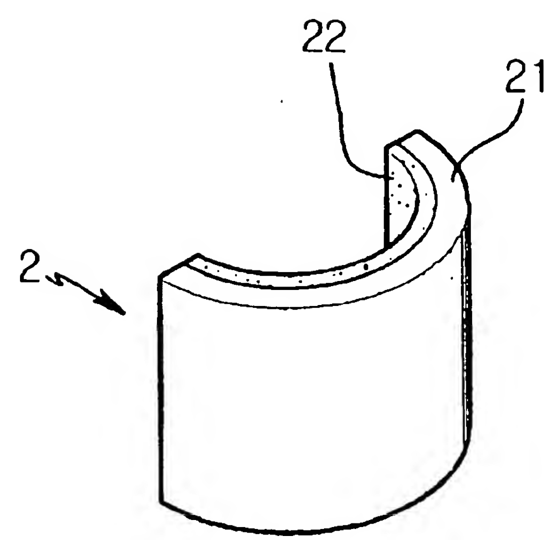


Fig. 3

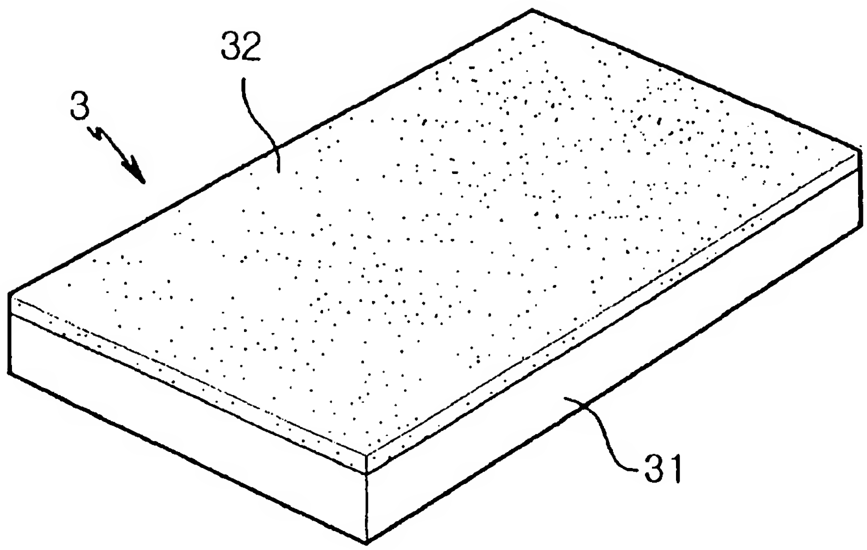


Fig. 4

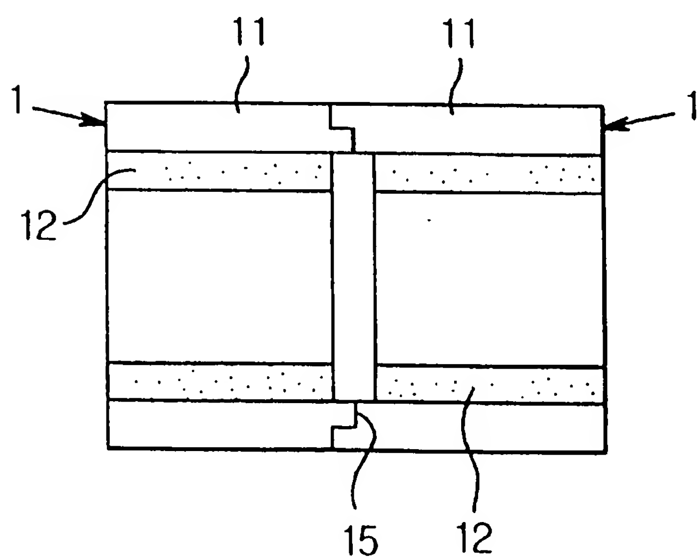


Fig. 5

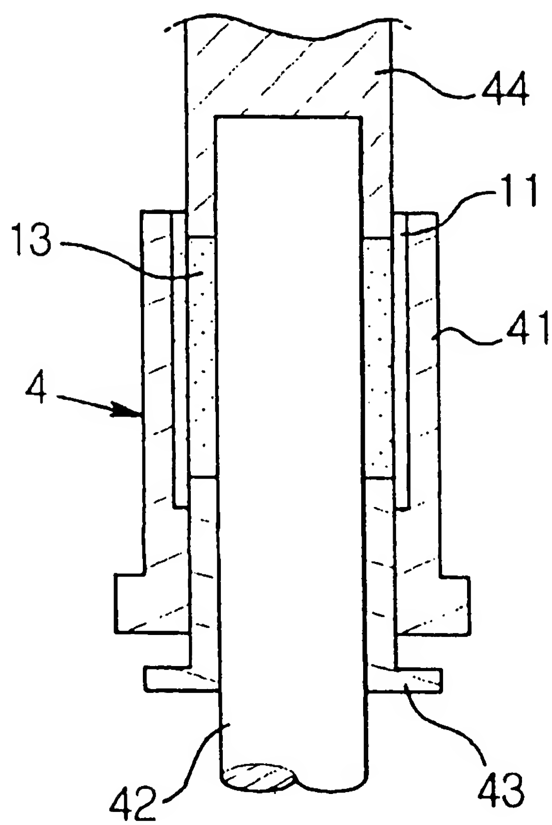


Fig. 6

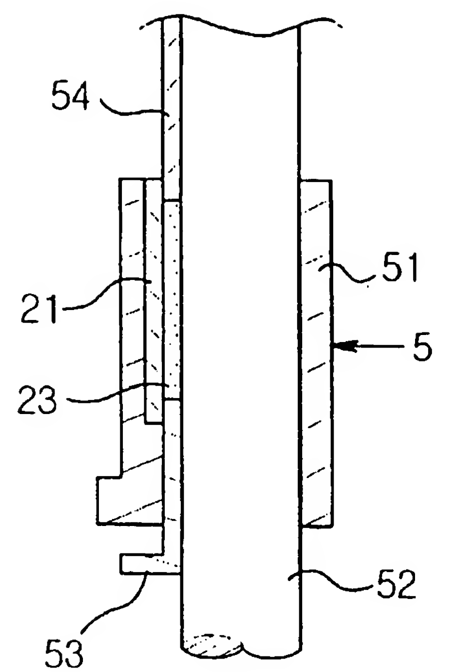


Fig. 7

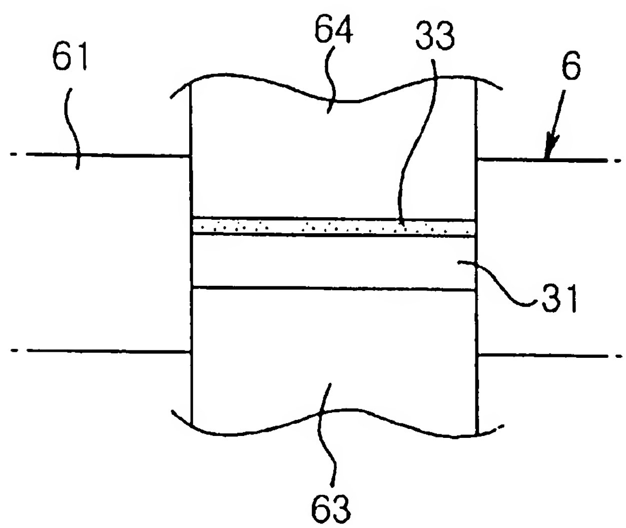


Fig. 8

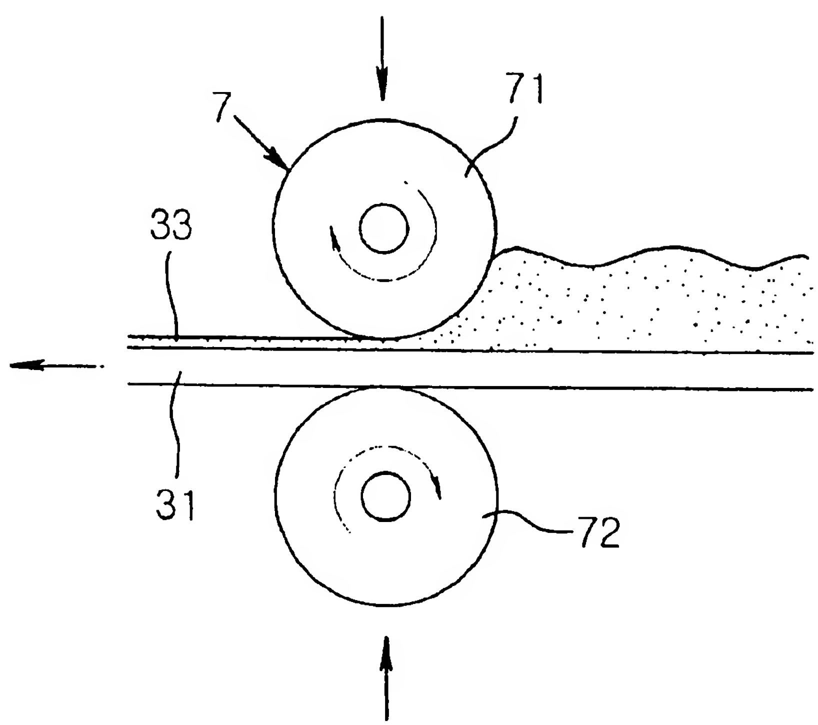


Fig. 9

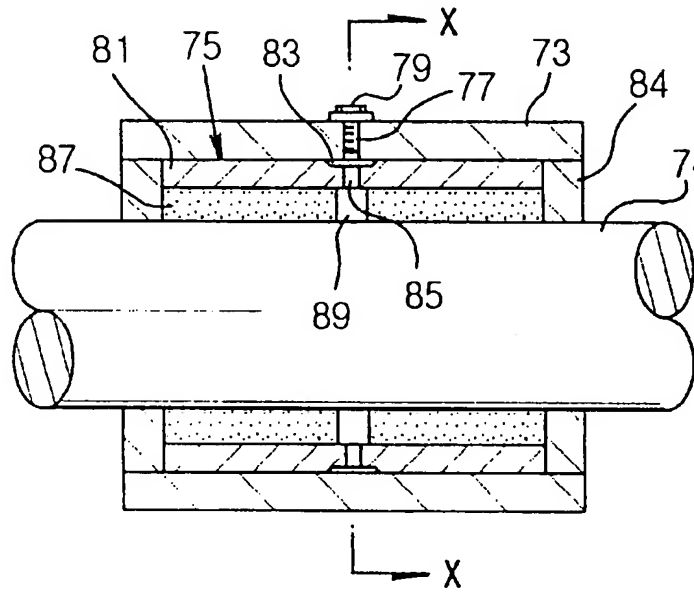


Fig. 10

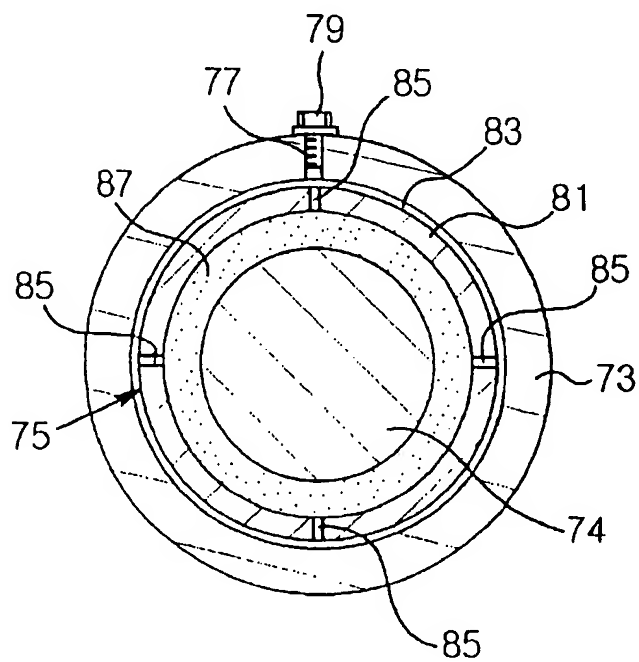
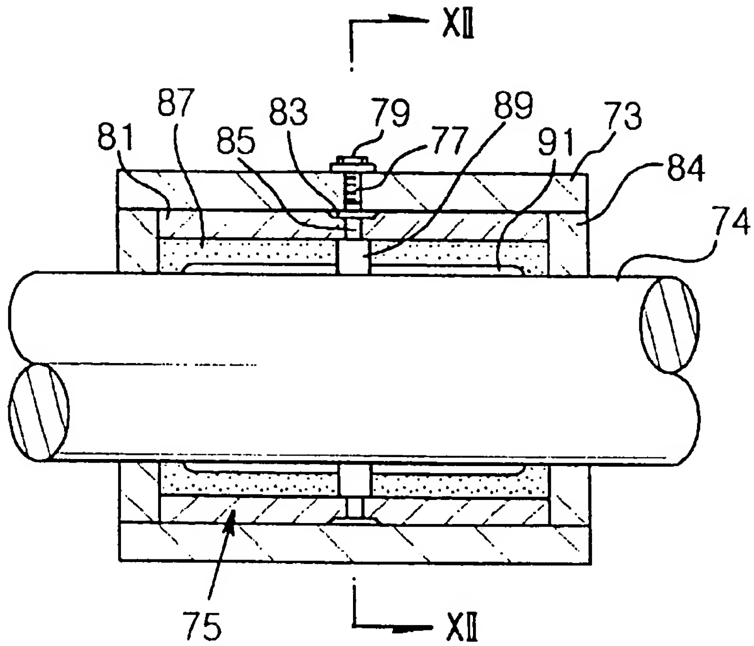


Fig. 11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XII

Fig. 12

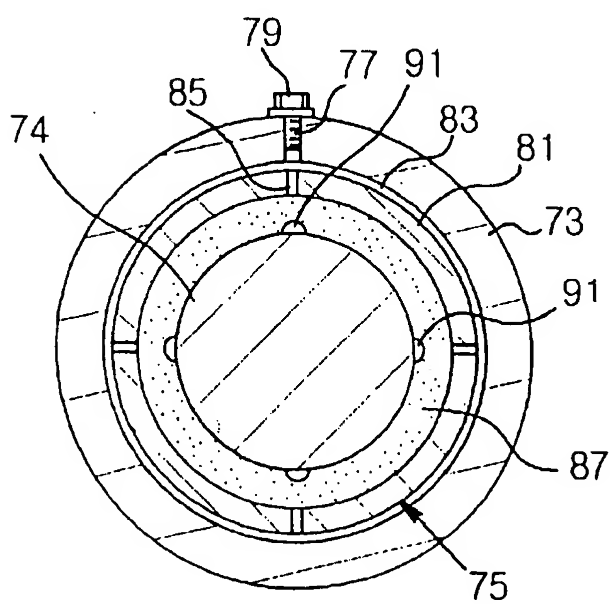
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Fig. 13

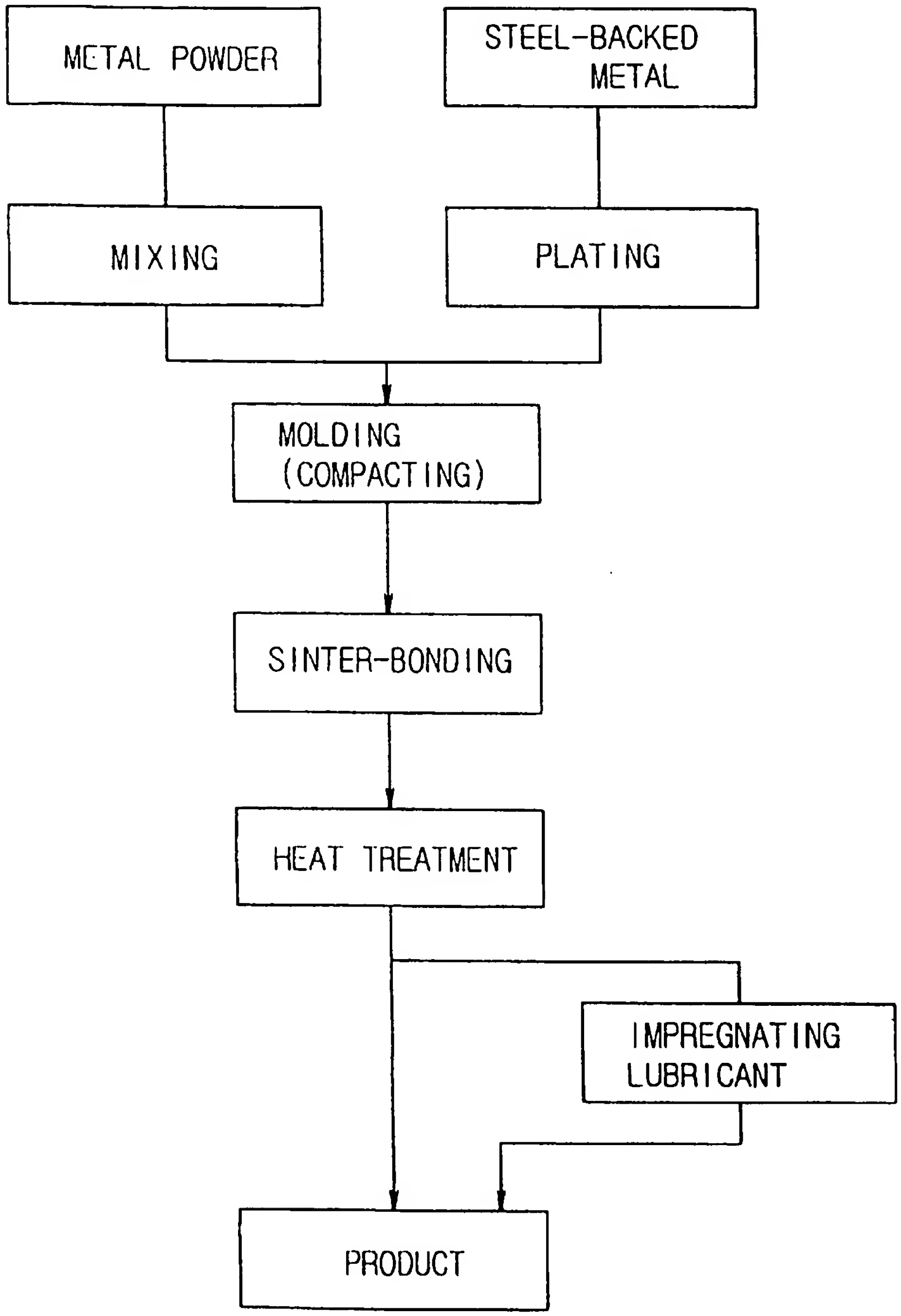
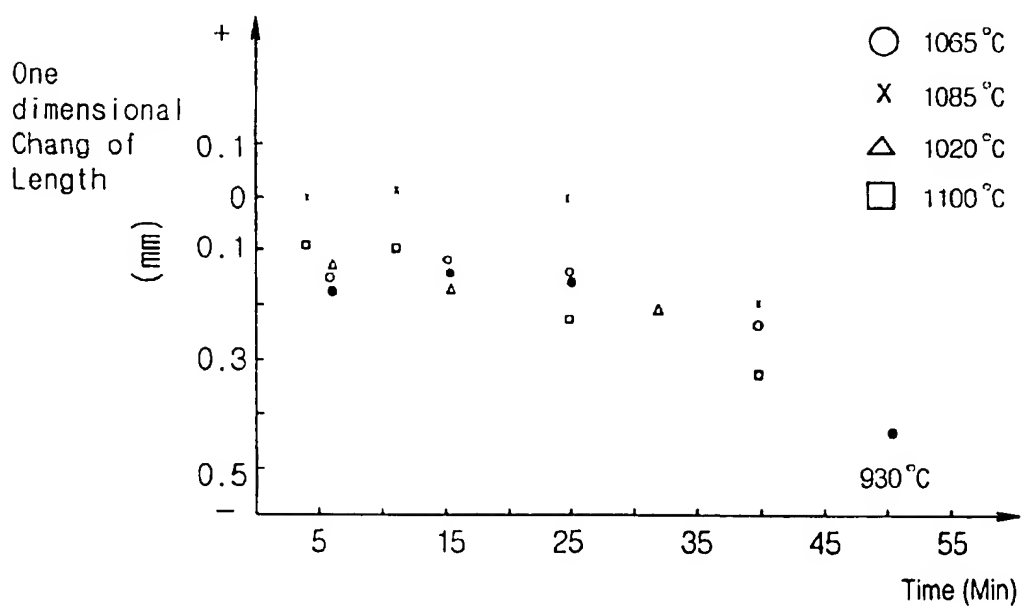

METAL POWDER
STEEL-BACKED METAL
MIXING
PLATING
MOLDING (COMPACTING)
SINTER-BONDING
HEAT TREATMENT
IMPREGNATING LUBRICANT
PRODUCT

Fig. 14


One dimensional Chang of Length (mm)
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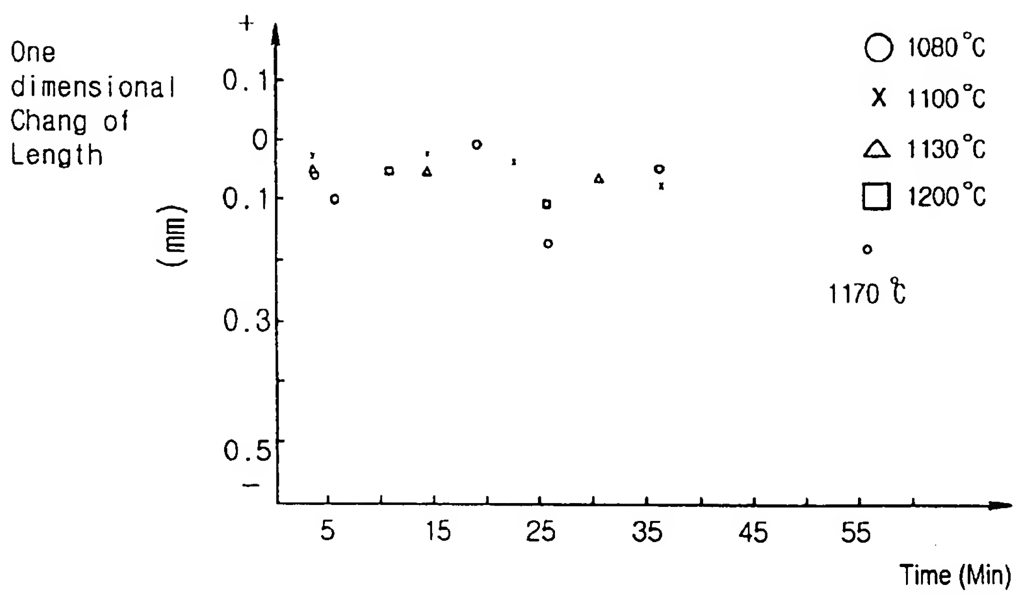
Time (Min)
1065°C
X 1085°C
1020°C
1100°C
930°C

Fig. 15


One dimensional Chang of Length
(mm)
+
0.1
0
0.1
0.3
0.5
−
5
15
25
35
45
55
Time (Min)
1080°C
1100°C
1130°C
1200°C
1170 ℃

SLIDING BEARING AND MANUFACTURING METHOD THEREOF

BACKGROUND OF THE INVENTION

1. Field of the Invention

The present invention relates to a sliding bearing and a manufacturing method thereof, and more particularly, to a sliding bearing having a sintered alloy layer formed on a steel-backed metal layer by sintering a compacted body made of iron-based metal mixture powder and concurrently bonding the same to the steel-backed metal layer so that load-carrying capacity (permissible load), fatigue strength, and fitting characteristic in the housing are improved, and also having lubricating oil meeting the optimal lubrication conditions impregnated therein so that a grease refilling period is considerably extended, facilitating maintenance of the sliding bearing, and a manufacturing method of the same.

2. Description of the Related Art

In general, a sliding bearing is a machine part for supporting a shaft, a journal, or a counter plate used for a driving or sliding portion of industrial machines such as construction equipments of excavator, forklift, crane, machine tools, presses, injection machines, or vehicles. The sliding bearing can be classified into various types such as a bush type, a half-pipe type, and a plate type, and usually formed by bonding copper-based alloy, aluminum-based alloy, copper-lead-based alloy, or copper-based and synthetic resin-based complex material to a steel-backed metal layer.

The structure of each conventional sliding bearing will be described as follows.

First, a sliding bearing formed by bonding an aluminum-based alloy layer to a steel-backed metal layer includes a steel-backed metal layer, an intermediate layer attached to the steel-backed metal layer, and an aluminum bearing alloy layer attached to the intermediate layer. The aluminum alloy layer includes a small amount of tin (Sn) and silicon (Si), and the intermediate layer is formed of an aluminum alloy layer including a small amount of one or more elements selected from the group consisting of manganese (Mn), copper (Cu), and magnesium (Mg).

Second, a sliding bearing having a copper-lead-based alloy layer has a multi-layered structure composed of a steel-backed metal layer, a copper-lead-based bearing alloy layer combined to the steel-backed metal layer, and a tin-containing lead alloy combined to the bearing alloy layer.

Third, there is a sliding bearing having a bearing layer formed of synthetic resin as a main ingredient which is formed on the surface thereof. In this sliding bearing, a porous metal layer is formed on an inner metal layer such as a steel plate and a bearing layer having lubricating synthetic resin as a main ingredient which covers a surface of the porous metal layer and part of which is impregnated in the pores of the porous metal layer. The bearing layer is formed by solidifying 4-ethylene resin fluoride and solid lubricant so that fine powder particles of the PTFE resin and the solid lubricant cohere each other.

However, the conventional sliding bearings having the above structures have demerits. That is, since the alloy layers contacting the counterpart of the sliding bearing are formed of copper-, copper-tin-, copper-lead-, aluminum-, or synthetic resin-based material exhibiting a relatively low hardness, it is not appropriate to use these sliding bearings for the permissible load over 300–500 kgf/cm^2. Also, since it is not possible to increase strength and hardness through heat treatment, wear resistance of the sliding bearing is improved.

To overcome the above problems, as a sliding bearing, a sintered alloy bearing has been used having only an iron-based sintered alloy layer without a steel-backed metal layer. However, the sintered alloy bearing is only used in a small size, i. e., the diameter and length of the bearing are within a range of 5–50 mm, since it is considerably difficult to make the large size sintered bearing have the high density. Furthermore, when solid lubricant is included in the sintered alloy bearing and the high pressure for molding (compacting) is applied to increase the density of the bearing, the solid lubricant acts as an impediment to the sintering so that the strength and load-carrying capacity of the bearing is further reduced. Thus, with the iron-based sintered alloy layer only, low friction and high load-carrying capacity (or high permissible load) cannot be simultaneously achieved.

To solve the above problems, a sliding bearing in which a steel-backed metal layer is bonded to an iron-based sintered alloy layer has been proposed. Here, the sintered alloy layer is required to include at least an iron component of 70 wt % or more and porosity of 10–30%, to satisfy conditions of a predetermined load-carrying capacity, high resistance to fatigue strength and fitting characteristic in the housing, prevention of detachment from the housing, and prevention of oil loss through a rear side. However, a sintered alloy layer satisfying such conditions is not easily overlaid onto the steel-backed metal layer since, when an iron-based mixture powder body of such composition is sintered, the mixture powder body severely shrinks due to densification phenomenon. Also, since the steel-backed metal layer expands at the corresponding sintering temperature, the iron-based sintered body and the steel-backed metal layer are not bonded to each other and thus the bearing is manufactured in a state the iron-based sintered body and the steel-backed metal layer being separated.

Due to the difficulties in bonding the steel-backed metal layer and the sintered alloy layer, in order to manufacture a sliding bearing of which an iron-based sintered alloy layer is bonded to a steel-backed metal layer, a sintered alloy layer is first formed by sintering iron-based metal alloy powder and then the sintered alloy layer is bonded to a steel-backed metal layer by a method such as welding or brazing. In this case, however, a process for manufacturing the sliding bearing is divided into two steps and further pores can be clogged by a filler metal fused during brazing which is easily absorbed in the pores of the sintered alloy layer due to capillary action. As a result, it is difficult to obtain porosity of 10% or more in volume in the sintered alloy layer for impregnation of the lubricant, and brittle structures are formed due to a reaction between the fused filler metal and the sintered alloy so that the sintered alloy layer is brittle against impacts. Also, flux used for brazing remains in the pores of the sintered alloy layer so that the sintered alloy layer can be easily and rapidly corroded, causing a bad effect of the lubricating property of the sliding bearing.

Meanwhile, for the best performance of the lubricant of the sliding bearing, the sintered alloy layer must be impregnated with lubricating oil. However, since the best conditions of impregnation of the lubricating oil for the sliding bearing having an iron-based sintered alloy layer sinter-bonded to a steel-backed metal layer has not been known so far, the performance of the lubricant can not be maximized.

Further, since grease must be frequently refilled as necessary after the conventional sliding bearing is initially

installed, the maintenance and management of the sliding bearing is difficult and inconvenient.

OBJECTS OF THE INVENTION

To solve the above problems, it is an objective of the present invention to provide a sliding bearing which can improve dynamic load-carrying capacity (permissible load) including impact load, wear resistance, fatigue strength, and fitting characteristic in the housing and also prevent detachment from housing and the loss of oil through a rear side.

It is another objective of the present invention to provide a sliding bearing in which a self-lubrication is performed smoothly and a dry/wet lubrication is performed altogether.

It is still another objective of the present invention to provide a sliding bearing impregnated with lubricating oil having the optimal viscosity and satisfying lubrication conditions for an iron-based sintered alloy layer.

It is yet another objective of the present invention to provide a sliding bearing having a grease refilling structure so that a frequent, manual replenishing of grease is not needed and a grease refilling period is extended, thereby accomplishing long term trouble-free maintenance.

It is yet further another objective of the present invention to provide a method of manufacturing a sliding bearing by which a sintered alloy layer is formed by sintering an iron-based metal mixture powder molded body (compacted body) and concurrently bonding the sintered body to a steel-backed metal layer.

SUMMARY OF THE INVENTION

Accordingly, to achieve the above objectives, there is provided a sliding bearing including a steel-backed metal layer, and an iron-based sintered alloy layer formed of copper of 10–30 wt % and iron for the balance. The iron-based sintered alloy layer is sinter-bonded (sintered and concurrently bonded) to the steel-backed metal layer.

It is preferable in the present invention that the iron-based sintered alloy layer further includes graphite of 0.1–6.5 wt % and molybdenum disulfide of 0.1–7.0 wt %.

To achieve the above objectives, there is provided a sliding bearing in which lubricating oil having ISO viscosity grade of 100–1,500, kinematic viscosity of 98–1,500 cSt at 40° C., and viscosity index of 120–50, or lubricating oil having ISO viscosity grade of 220–680, kinematic viscosity of 210–670 cSt at 40° C., and viscosity index of 90–110, is impregnated into the iron-based sintered alloy layer.

To achieve the above objectives, there is provided a sliding bearing including a housing having an inlet capable of opening or being closed formed on the outer surface of the steel-backed metal layer, for supporting the sliding bearing, an annular guide groove formed along the outer circumferential surface of the steel-backed metal layer at the position corresponding to the inlet, a plurality of first grease supply holes radially penetrating the steel-backed metal layer along the guide groove, and a second grease supply groove (reservoir) connecting to the first grease supply holes circumferentially extending in the ring shape (annular type) in said iron-based sintered alloy layer. Therefore, the grease injected via the inlet of the housing flows along the guide groove along the outer circumferential surface of the steel-backed metal layer and passes through the first grease supply holes and stored in the second grease supply groove, and finally is supplied to the boundary surface between the sintered alloy layer and the sliding counterpart.

To achieve the above objectives, there is provided a method of manufacturing a bush type sliding bearing by inserting a steel-backed metal layer into a mold of a press having cavity for installation of a core at the central portion of the mold and a lower pressing member at the lower portion thereof; filling metal mixture powder formed of copper of 10–30 wt %, graphite of 6.5 wt % or less, and iron for the balance between the core and the steel-backed metal layer and then inserting an upper pressing member into the upper portion of the mold; forming the metal mixture powder into a molded body by selectively applying pressure of 50–300 kgf/cm^2 to the upper pressing member and the lower pressing member, respectively; and sintering the molded body into a sintered alloy layer by maintaining the molded body together with the steel-backed metal layer under predetermined gas atmosphere at temperature of 1,065° C.–1,095° C. for 3–25 minutes and concurrently bonding the sintered alloy layer to the steel-backed metal layer.

BRIEF DESCRIPTION OF THE DRAWINGS

The above objectives and advantages of the present invention will become more apparent by describing in detail a preferred embodiment thereof with reference to the attached drawings in which:

FIG. 1 is a perspective view illustrating a bush type sliding bearing according to the present invention;

FIG. 2 is a perspective view illustrating a half-pipe type sliding bearing according to the present invention;

FIG. 3 is a perspective view illustrating a plate type sliding bearing according to the present invention;

FIG. 4 is a sectional view showing a plurality of the bush type sliding bearings of a diameter as shown in FIG. 1 are welded or bonded together to be lengthy;

FIG. 5 is a sectional view schematically showing a press for manufacturing the sliding bearing shown in FIG. 1;

FIG. 6 is a sectional view schematically showing a press for manufacturing the sliding bearing shown in FIG. 2;

FIG. 7 is a sectional view schematically showing a press for manufacturing the sliding bearing shown in FIG. 3;

FIG. 8 is a sectional view schematically showing a rolling mill for manufacturing the sliding bearing shown in FIG. 3;

FIG. 9 is a sectional view showing the structure of a grease supply of the sliding bearing shown in FIG. 1;

FIG. 10 is a sectional view taken along line X—X of FIG. 9;

FIG. 11 is a sectional view showing the structure of a grease supply of a sliding bearing according to another preferred embodiment of the present invention;

FIG. 12 is a sectional view taken along line XII—XII of FIG. 11;

FIG. 13 is a flow chart representing a process of manufacturing a sliding bearing according to the present invention;

FIG. 14 is a graph showing one dimensional change in length with respect to the sintering temperature according to the first preferred embodiment of the present invention; and

FIG. 15 is a graph showing one dimensional change in length with respect to the sintering temperature according to the second preferred embodiment of the present invention.

DETAILED DESCRIPTION OF THE INVENTION

A sliding bearing according to the present invention can have various shapes according to the purpose of use, for

example, a bush type, a half-pipe type, and a plate type as respectively shown in FIGS. 1 to 3.

Referring to FIGS. 1 and 2, in a bush type sliding bearing 1 and a half-pipe type sliding bearing 2, steel-backed metal layers 11 and 21 are disposed at the outer circumferential surfaces of the respective bearings 1 and 2, and iron-based metal mixture powder is sinter-bonded (concurrently sintered and bonded) to the inner circumferential surface of each steel-backed metal layer. Thus, iron-based sintered alloy layers 12 and 22 are formed on the inner circumferential surfaces of the steel-backed metal layers 11 and 21, respectively. In a plate type sliding bearing 3, as shown in FIG. 3, iron-based metal mixture powder is concurrently sintered and bonded to a surface of a steel-backed metal layer 31 and thus an iron-based sintered alloy layer 32 is formed on the surface of the steel-backed metal layer 31.

According to the preferred embodiment of the present invention, the sintered alloy layers 12, 22, and 32 are preferably comprising of copper (Cu) of 10–30 wt % and iron (Fe) for the residue. Here, graphite of 6.5 wt % or less may be further added to the sintered alloy layers 12, 22, and 32.

Also, according to the present invention, the sintered alloy layers 12, 22, and 32 can further include molybdenum disulfide (MoS_2), tin (Sn), nickel (Ni), and manganese (Mg) in addition to iron, copper, and graphite. Here, the composition ratio is preferably copper of 10–30 wt %, graphite of 6.5 wt % or less, molybdenum disulfide of 0.1–7.0 wt %, tin of 2–15 wt %, nickel of 2–15 wt %, manganese of 2–5 wt %, and iron for the residue.

Since the sliding bearings 1, 2, and 3 have the iron-based alloy layers 12, 22, and 32, respectively, a heat treatment is possible to increase strength and hardness.

The graphite or molybdenum disulfide included in the sintered alloy layers 12, 22, and 32 serves as a solid lubricant. However, it is preferable that a liquid lubricant such as lubricating oil is impregnated into the sintered alloy layers 12, 22, and 32 to meet a long-term grease refilling condition.

As the lubricant oil impregnated into the sliding bearing according to the present invention, for a sliding bearing of a general use, gear oil meeting the conditions of ISO viscosity grade of 100–1,500, kinematic viscosity of 98–1,500 cSt at 40° C., and viscosity index of 120–50 is used according to conditions such as applied load and velocity of a sliding counterpart.

To meet the conditions of heavy permissible load over 300–500 kgf/cm^2 and velocity over 5–70 mm/sec, it is preferable to use lubricating oil having ISO viscosity grade of 220–680, kinematic viscosity of 210–670 cSt at 40° C., and viscosity index of 90–110. When the viscosity of lubricating oil is within the above ranges, a frictional coefficient obtained when the sliding counterpart is a chromium-plated hardened carbon steel becomes quite a low value in the range of 0.05–0.15.

In particular, to provide the lubricating oil used for heavy load with extreme-pressure resisting characteristic in addition to the lubrication characteristic, it is preferable that graphite and molybdenum disulfide powders having the grain size less than 200 μm serving as a solid lubricant, are selectively or altogether added up to 50% in volume ratio as being suspended into the lubricating oil. When the lubricating oil having graphite and molybdenum disulfide powders added in addition to the wet lubricant such as lubricating oil is used, the frictional coefficient obtained in case that the sliding counterpart of chromium-plated hardened carbon steel is 0.07 or less, indicating a stable lubricating performance under application of the extremely-high static pressure and dynamic loads. When graphite and molybdenum disulfide powders are mixed with the liquid lubricant such as lubricating oil, the powders must be completely mixed to be used until they are suspended or reach a paste state.

Meanwhile, the steel-backed metal layer of the present invention is formed of carbon steel, and preferably, is copper-plated to the thickness of 2–10 μm by electroless plating or electroplating. Here, stainless steel may be used for the steel-backed metal layer.

Also, according to the present invention, as shown in FIG. 4, a lengthy sliding bearing can be made by welding or bonding a plurality of sliding bearings in a lengthwise direction.

A sliding bearing is installed in a housing to support the sliding counterpart such as a shaft or a journal, and grease is supplied to the sliding bearing.

A sliding bearing having a grease supply structure according to an example of the present invention will be described with reference to the drawings.

Referring to FIGS. 9 and 10, a sliding bearing 75 in which an iron-based sintered alloy layer 87 is bonded to the inner circumferential surface of a steel-backed metal layer 81 is installed inside a housing 73 for supporting a sliding counterpart 74 such as a shaft or a journal, and a dust seal 84 is further installed to both sides of the sliding bearing 75. An inlet 77 for a lubricant is formed at upper side of the housing 73 and a nipple 79 for opening/closing the inlet 77 is coupled to the inlet. An annular guide groove 83 for guiding a lubricant such as grease injected through the inlet 77 toward the outer circumferential surface of the steel-backed metal layer 81 is formed at the outer circumferential surface of the steel-backed metal layer. A plurality of first grease supply holes 85 for guiding the grease injected along the guide groove 83 to the sintered alloy layer 87 are formed along the guide groove. In the present embodiment, as shown in FIG. 10, four first grease supply holes 85 are respectively formed in four perpendicular directions.

A second grease supply groove 89 connecting to the first grease supply holes 85 for guiding the grease flowing via the first grease supply holes 85 to reach a boundary surface between the sintered alloy layer 87 and the sliding counterpart 74, are formed toward the circumferential direction. The second grease supply groove 89 is a ring-shaped or annular type groove formed by removing the sintered alloy layer 87 along the circumferential direction thereof. The width of guide groove 83 and second grease supply groove 89 must be greater than the diameter of each first grease supply hole 85, while the second grease supply groove 89 has no limit in its width. Also, the height of second grease supply groove 85 equals to the thickness of the sintered alloy layer 87.

Thus, when the nipple 79 is detached from the inlet 77 to inject grease, the grease is supplied along the guide groove 83 toward the outer circumferential surface of the steel-backed metal layer 81 and stored in the second grease supply groove 89 passing through the first grease supply holes 85. Thus, the grease is supplied to the boundary surface between the sintered alloy layer 87 and the sliding counterpart 74 whenever the sliding bearing 75 operates. Since the grease supplied to the boundary surface includes graphite or molybdenum disulfide, selectively or altogether, lubrication operations are achieved very effectively due to the combined operation of dry lubrication and wet lubrication, i.e., operations of the solid lubricant, the impregnated oil, and the grease.

The dry lubrication operation by a solid lubricant, graphite or molybdenum disulfide, shows better performance for supporting the sliding counterpart **74** of heavy load which moves at low velocity. Also, since the solid lubricant stored in the second grease supply groove **89** and at the boundary surface hardly leaks or is reduced, a period for refilling the grease is extended for a long term.

When the above-described grease supply structure is applied to a case in which the sliding counterpart **74** is formed of chromium-plated hardened carbon steel, the frictional coefficient between the sliding bearing **75** and the sliding counterpart is maintained within a range of 0.02–0.05, thus considerably reducing the friction therebetween. Further, since most solid lubricant remains in the second grease supply groove **89** and at the boundary surface, together with the grease, only a wet lubricant, e. g., lubricating oil or grease, suffices for refilling, without supplying a solid lubricant.

Referring to FIGS. **11** and **12**, a sliding bearing having a grease supply structure according to another example of the present invention will be described.

Since a grease supply structure according to another example of the present invention has the same structure and operation as the above-described grease supply structure example, except that diffuse grooves **91** are formed on the inner circumferential surface of the sintered alloy layer **87** in an axial direction extending from the second grease supply groove **89**, the detailed description thereof will be omitted.

Although the diffuse grooves **91** can be formed into various shapes, it is preferred that the profile of each diffuse groove is half-circular. The grease injected through the inlet **77** flows on the circumferential surface of the steel-backed metal layer **81** along the guide groove **83** and stored in the first grease supply holes **85** and second grease supply groove **89**. The stored grease finally flows into the diffuse grooves **91** as the sliding bearing **75** operates. Thus, since the diffuse grooves **91** are formed axially on the inner circumferential surface of the sintered alloy layer **87** and the grease is stored in the diffuse grooves **91**, the lubrication operation to the boundary surface between the sintered alloy layer **87** and the sliding counterpart **74** is performed very effectively. Also, since more amount of the lubricant remains on the boundary surface than in the case according to the first example due to the amount stored in the diffuse grooves **91**, the period for refilling grease can be further extended.

A method for manufacturing a sliding bearing having the above structure according to the present invention will now be described with reference to the drawings.

Referring to FIGS. **1**, **4**, **5**, and **13**, a method of manufacturing a bush type sliding bearing will be first described. In this method, copper-plated steel-backed metal is used for the steel-backed metal layer **11**, and preferably, a copper plate layer (not shown) is formed to have a thickness of 2–10 μm by electroless plating or electroplating.

Then, metal mixture powder having the above composition ratio is uniformly mixed. Next, the steel-backed metal layer **11** is inserted in a mold **41** of a press **4**, in which a core **42** and a lower pressing member **43** are installed at the central and lower portions of the press, respectively. The mixed metal mixture powder is filled between the steel-backed metal layer **11** and the core **42**, and an upper pressing member **44** inserts into the upper portion of the mold **41**.

Then, a molded body (compacted body) **13** is formed by press-molding the metal mixture powder and then the molded body and the steel-backed metal layer **11** are taken out together. In the press-molding, the upper and lower pressing members **44** and **43** may operate selectively or altogether. The pressure during pressing is preferably maintained between 50–300 kg/cm2 for the respective upper and lower pressures, and the molding pressure is determined in consideration of a desired porosity (the volume ratio of pores with respect to the volume of the entire sintered alloy layer) after sinter-bonding (sintering and bonding) processes which will be described later. It is preferable that the porosity is between 10–30%.

Next, the molded body **13** is heated together with the steel-backed metal layer **11** and is sintered and bonded each other so that the iron-based sintered alloy layer **12** is formed on the steel-backed metal layer **11**. The sintering/bonding process is performed under any one gas atmosphere selected from the group consisting of nitrogen, hydrogen, and mixed gas of nitrogen and hydrogen atmospheres, ammonia decomposed gas atmosphere, argon gas atmosphere, vacuum, endothermic and exothermic gas atmosphere. In the nitrogen and hydrogen mixed gas atmosphere, hydrogen is preferably included over 30% in volume ratio with respect to the mixed gas. Also, in the vacuum atmosphere, it is preferable the pressure is 10^{-2} torr or less. Further, the sintering is performed at the temperature between 1,065° C.–1,095° C. and the temperature is kept for 3–25 minutes.

When the sintering temperature and time is out of the above ranges, the molded body **13** can be sintered, but not bonded well to the steel-backed metal layer **11**. When the molded body **13** is sintered and turned into the sintered alloy layer **12** in the above ranges set for sintering temperature and time, contraction hardly occurs or only a little expansion is generated so that the sintered alloy layer **12** is easily bonded to the steel-backed metal layer **11**. However, since liquid-state sintering occurs when the sintering is made above the set ranges for temperature and time while solid-state sintering occurs under the above set ranges, the sintered alloy layer **12** severely contracts and is detached from the steel-backed metal layer **11**, making the bonding of the sintered alloy layer and the steel-backed metal difficult. Also, when solid sintering is occurred, strength of the sintered alloy layer **12** is weakened. When the liquid sintering is occurred, copper is fused during the sintering process and fused copper exhibiting high fluidity is formed which flows on the boundary surface between the steel-backed metal layer **11** and the sintered alloy layer **12**, or concentrates on any one position, so that pores is irregularly distributed throughout the sintered alloy layer **12**, causing ill effect on porosity. Further, as fluidity of the fused copper becomes higher, the contraction rate of the sintered alloy layer **12** becomes higher.

When sintering is made within transition temperature and transition time between each solid-state and liquid-state sintering, copper included in the molded body **13** is fused and fused copper is formed. However, since viscosity of the above fused copper is relatively high, i. e., fluidity is low, the concentration phenomenon that the fused copper flows toward the boundary surface or at any one position thereof can be prevented, thereby making the distribution of the fused copper uniform. Therefore, a degree of clogging of the pores in the sintered alloy layer **12** is reduced so that the pores can be uniformly distributed. Also, owing to the fused copper of high viscosity present between the sintered alloy layer **12** and the steel-backed metal layer **11**, the sintered alloy layer is sintered and concurrently bonded (sinter-bonded) together.

The iron-based metal mixture powder not satisfying the above composition ratio is not bonded well, concurrently with sintering thereof. In such a case, a change of the copper content serving as a critical factor in sintering is preferred.

Then, the sliding bearing obtained after sintering and bonding undergoes heat treatment such as carburizing, nitriding, carburizing and nitriding, quenching, or tempering after these treatments, to improve its strength and hardness. These heat treatments are possible since the sintered alloy layer **12** is formed using iron as a main component. When the heat treatment is performed in a continuous furnace, it is preferable that the sliding bearing **1** is transferred from a heating place to a cooling place and then maintained at temperature between 750° C.–950° C. for performing carburizing, nitriding, carburizing and nitriding, quenching, or tempering.

When graphite or molybdenum disulfide is included in the sintered alloy layer **12**, it serves as a dry lubricant. Further a step of impregnating a wet lubricant such as lubricating oil into the heat-treated sliding bearing **1** can be included. Preferably, the liquid lubricant used in the impregnating step is lubricating oil having ISO viscosity grade of 100–1,500, kinematic viscosity of 98–1,500 cSt at 40° C., and viscosity index of 120–50, or lubricating oil having ISO viscosity grade of 220–680, kinematic viscosity of 210–670 cSt at 40° C., and viscosity index of 90–110.

Next, a method of manufacturing the half-pipe type sliding bearing **2** will be described with reference to FIGS. **2** and **6**.

The steel-backed metal layer **21** is inserted into a mold **51** of a press **5** which has a half-circular cavity, a core **52** installed at the center thereof, and a lower pressing member **53** installed at the lower portion thereof. Then, iron-based metal mixture powder is filled between the steel-backed metal layer **21** and the mold **51**, and an upper pressing member **54** is inserted into the upper portion of the mold **51** and presses the iron-based metal mixture powder to form a molded body **23**. The molded body **23** is sintered together with the steel-backed metal **21** under the same conditions as described above so that the sliding bearing **2** having the sintered alloy layer **22** formed on the steel-backed metal layer may be manufactured. Since the other steps are the same as in the method for manufacturing a bush type sliding bearing, a detailed description thereof is omitted. Also, by dividing the bush type sliding bearing **1** into two halves, two half-pipe type sliding bearings can be obtained.

Next, a method of manufacturing the plate type sliding bearing **3** will be described, with reference to FIGS. **3**, **7**, and **8**.

In this method, metal mixture powder having the above composition ratio is sintered to the copper-plated steel-backed metal layer **31** of a plate type to a predetermined thickness and concurrently bonded to form the iron-based sintered alloy layer **32**. As shown in FIG. **7**, the steel-backed metal layer **31** is placed on a mold (die) **61** of a press **6** having a rectangular cavity where a lower pressing member **63** is inserted in the lower portion thereof.

Then, metal mixture powder having the above composition ratio is placed on the steel-backed metal layer **31** and an upper pressing member **64** is inserted into the upper portion of the mold **61**. The upper and lower pressing members **64** and **63** press the metal mixture powder to form a molded body **33**.

Then, the molded body **33**, together with the steel-backed metal layer **31**, is kept under a predetermined gas atmosphere at predetermined temperature for predetermined time as above-described so that the molded body **33** is sintered and concurrently bonded to the steel-backed metal layer **31** to form the sintered alloy layer **32**.

Alternatively, as shown in FIG. **8**, a rolling mill **7** can be used, that is, the steel-backed metal layer **31** is placed on the rolling mill **7** and the metal mixture powder of the above composition ratio is placed thereon.

Then, the steel-backed metal layer **31** having the metal mixture powder placed thereon passes between upper and lower pressing rollers **71** and **72** and is maintained under a predetermined gas atmosphere at predetermined temperature for predetermined time as described-above. Thus, a molded body is sintered and concurrently bonded to the steel-backed metal layer **31**, to form the iron-based alloy layer **32**. If necessary, a heat treatment process or a lubricant impregnating step is applied as in the method for manufacturing the bush type sliding bearing.

In the method of manufacturing the plate type sliding bearing **3**, the sintering temperature and the maintenance time are preferably in ranges of 1,065° C.–1,250° C. and 3–35 minutes, respectively, unlike the bush type or half-pipe type sliding bearing manufacturing method. Under such conditions, defective bonding does not occur caused by the difference in relative contraction rate and thermal expansion between the sintered alloy layer **32** and the steel-backed metal layer **31**. Also, the defective bonding does not occur when the metal mixture powder satisfies the condition of the above-described composition ratio. If the condition is not met, bonding is not made as desired. That is, when a sintering/bonding (i. e. sinter-bonding) process is performed out of the above condition, in the plate type sliding bearing **3**, unlike the bush type sliding bearing **1**, the middle portion of the boundary surface is bonded while defectiveness that the edge portion thereof is debonded or the entire surface is not bonded occurs. The latter defectiveness occurs due to the composition ratio of the metal mixture powder, as described above.

Preferred Embodiment 1

Metal mixture powder consisting of iron of 74 wt %, copper of 25 wt %, and graphite of 1 wt % is pressed by upper and lower pressure of respectively 50–300 kgf/cm^2 so that a molded body of 94 mm in inner diameter and 4 mm in thickness is formed. The molded body is maintained together with a steel-backed metal layer of 102 mm in inner diameter and 6 mm in thickness at the sintering temperature of 930° C.–1,100° C. for 5–60 minutes, and then is cooled, and thus a sliding bearing is manufactured. The contraction rate of the sintered alloy layer of the sliding bearing and a degree of bonding between the sintered alloy layer and the steel-backed metal layer are shown in a graph of FIG. **14**.

As shown in the graph, it can be seen that, when the conditions of sintering temperature between 1,065° C.–1, 095° C. and sintering time between 3–25 minutes are kept, no contraction occurs and further bonding is made well, concurrently with sintering thereof.

Preferred Embodiment 2

Metal mixture powder mixed with iron powder of 75 wt % and copper powder of 25 wt %, and metal mixture powder mixed of iron powder of 74 wt % and copper powder 25 wt %, and graphite powder 1 wt %, are formed into a molded body of a rectangular block having a size of 150 mm in length, 50 mm in width, and 20 mm in thickness by a press using a roll-press method. The molded body is maintained together with a copper-plated steel-backed metal layer under nitrogen and hydrogen mixture gas atmosphere at the temperature between 900° C.–1,200° C. for 5–60 minutes, and then is cooled, and thus a plate type sliding bearing is manufactured. The contraction rate of the sintered alloy layer of the sliding bearing and a degree of bonding between

the sintered alloy layer and the steel-backed metal layer are shown in a graph of FIG. 15. The molding pressure of 50–300 kg/cm^2 is applied as the respective upper and lower pressures.

As shown in the graph, it can be seen that the sintering temperature and the sintering time to realize a sintering/bonding state in which defective bonding due to relative contraction and expansion rate of the sintered alloy layer do not occur are between 1,065° C.–1,250° C. and between 3–20 minutes, respectively.

As described above, the sliding bearing and its manufacturing method according to the present invention have advantages as follows.

1) Since iron-based metal mixture powder molded in contact with a steel-backed metal layer is sintered in a transition range for solid-state sintering and liquid-state sintering, the iron-based metal mixture powder is sintered into an iron-based sintered alloy layer and concurrently and easily bonded to the steel-backed metal layer.

2) Since a strength-adjustable carbon steel is selected as a back metal layer and a low-density iron-based sintered alloy having lubrication and low friction characteristics is sinter-bonded to the inner surface of the back metal layer, the sliding bearing can support load over 700 kgf/cm^2. Thus, the sliding bearing according to the present invention shows superior performance not only under light load and high speed conditions, but also heavy load and medium or low speed conditions.

3) When the counterpart is made of chromium-plated hardened carbon steel, kinematic frictional coefficient is maintained between 0.05–0.15 and operation temperature are maintained under 30° C.–100° C.

4) When the housing is formed of iron-based, the sliding bearing according to the present invention can be easily fitted into the housing without any troubles and does not detach from the housing even under the severe change of atmospheric temperature since it has the elastic modulus almost identical to that of the housing and a low frictional coefficient

5) There is no frictional noise during reciprocation, oscillation and rotation in contact with a shaft.

6) The sliding bearing according to the present invention shows very excellent embeddability against foreign particles or wear debris.

7) The sliding bearing according to the present invention shows excellent resistance against crack-propagation, sudden impact load and a higher fatigue limit.

8) Seizure or burning of the inner surface of sintering material with respect to the counterpart does not occur in any event during operation.

9) The sliding bearing according to the present invention shows superior conformability and compatibility with the counterpart. Thus, particularly when the counterpart is formed of hardened chromium-plated carbon steel, it is hardly damaged.

10) There is no loss of oil through the rear side of the bearing due to presence of the steel-backed metal layer. Also, since the steel-backed metal layer is present on the rear surface of the bearing, recovery rate of an lubricating oil after the bearing operates is high due to a high hydrodynamic pressure to the pores near the back metal portion compared to a bearing having no back metal layer and a general-use bearing.

11) Since the rear surface of the bearing is formed of carbon steel, welding and bonding is made possible. Also,

since the width of the back metal layer of the bearing can be adjusted, the bearing itself can be formed to be a housing, or other supporters can be welded and bonded so that flexibility in design is allowed.

12) Since lubricating oil meeting the most'optimal conditions is impregnated into the sliding bearing, graphite or molybdenum disulfide included in the sintered alloy layer of the sliding bearing serves as a solid lubricant and also the lubricating oil impregnated into the sliding bearing serves as liquid lubricant. Further, since the impregnated lubricating oil includes graphite or molybdenum disulfide, selectively or altogether, the sliding bearing of the present invention always operates in a combined self-lubricated condition, the frictional coefficient with respect to the counterpart is very low, equivalent to the value of 0.05 or less, and the frictional coefficient is stably maintained in spite of the conditions of the extremely high static pressure and dynamic loads, thereby improving lubrication characteristic.

13) According to the grease supply structure of the present invention, since grease is smoothly supplied to the boundary surface between the sintered alloy layer and the sliding counterpart and remains in a large amount, the lubrication characteristic is superior and also an effect of combined self-lubricating operation increases the grease refilling period such that the grease refilling period is extended up to the minimum 300 hours to 1,000 hours. The grease refill period can be nearly permanent according to a method of designing the bearing. Further, even when grease refilling is needed, graphite or molybdenum disulfide which is a solid lubricant hardly leak out and remains in the inner ring-shape groove, so that the bearing can be maintained by refilling only grease.

What is claimed is:

1. A sliding bearing comprising:

a steel-backed metal layer; and

an iron-based sintered alloy layer comprising copper of 10–30 wt %, a solid lubricant and iron for the residue, the solid lubricant comprising at least one of graphite 0.1–6.5 wt % and molybdenum disulfide 0.1–7.0 wt %, said iron-based sintered alloy layer being sintered and concurrently bonded to said steel-backed metal layer at a temperature of 1,065–1095° C.

2. The sliding bearing as claimed in claim 1, wherein said iron-based sintered alloy layer further comprises tin of 2–15 wt %, nickel of 2–15 wt %, or manganese of 2–5 wt %.

3. The sliding bearing as claimed in claim 1, wherein the iron-based sintered alloy layer further comprises liquid lubricant impregnated therein.

4. The sliding bearing as claimed in claim 3, wherein the liquid lubricant comprises a lubricating oil having ISO viscosity grade of 100–1,500, kinematic viscosity of 98–1,500 cSt at 40° C., and viscosity index of 120–50, or a lubricating oil having ISO viscosity grade of 220–680, kinematic viscosity of 210–670 cSt at 40° C., and viscosity index of 90–110.

5. The sliding bearing as claimed in claim 3, wherein powders of at least one of graphite and molybdenum disulfide are suspended in the liquid lubricant, the powders having grain size less than 200 μm.

6. The sliding bearing as claimed in claim 1, wherein said sliding bearing further comprises:

a housing having an inlet formed on an outer surface of said steel-backed metal layer for supporting said sliding bearing;

an annular guide groove formed along an outer circumferential surface of said steel-backed metal layer at a position corresponding to said inlet;

a plurality of holes radially penetrating said steel-backed metal layer along said guide groove;

liquid lubricant impregnated into said iron-based sintered alloy layer, said liquid lubricant having ISO viscosity grade of 100–1,500, kinematic viscosity of 98–1,500 cSt at 40° C., and viscosity index of 120–50; and

powders of at least one of grahite and molybdenum disulfide, the powders being suspended in the liquid lubricant and having grain size less than 200 μm.

7. The sliding bearing as defined in claim 1, wherein the sintering of the alloy layer and the concurrent bonding at the temperature of 1065–1095° C. is performed for 3–25 minutes.

8. The sliding bearing as defined in claim 1, wherein before the sintering, a mixture comprising the copper, the solid lubricant and the iron is compacted under the pressure of 50–300 kg/cm^2.

* * * * *

US005545014A

United States Patent [19]

Sundberg et al.

[11] **Patent Number: 5,545,014**

[45] **Date of Patent: Aug. 13, 1996**

[54] **VARIABLE DISPLACEMENT VANE PUMP, COMPONENT PARTS AND METHOD**

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[73] Assignee: **Coltec Industries Inc.**, New York, N.Y.

[21] Appl. No.: **114,253**

[22] Filed: **Aug. 30, 1993**

[51] **Int. Cl.**[6] **F04B 23/10**

[52] **U.S. Cl.** **417/204**; 418/26; 418/268

[58] **Field of Search** 417/204; 418/26, 418/30, 268

[56] **References Cited**

U.S. PATENT DOCUMENTS

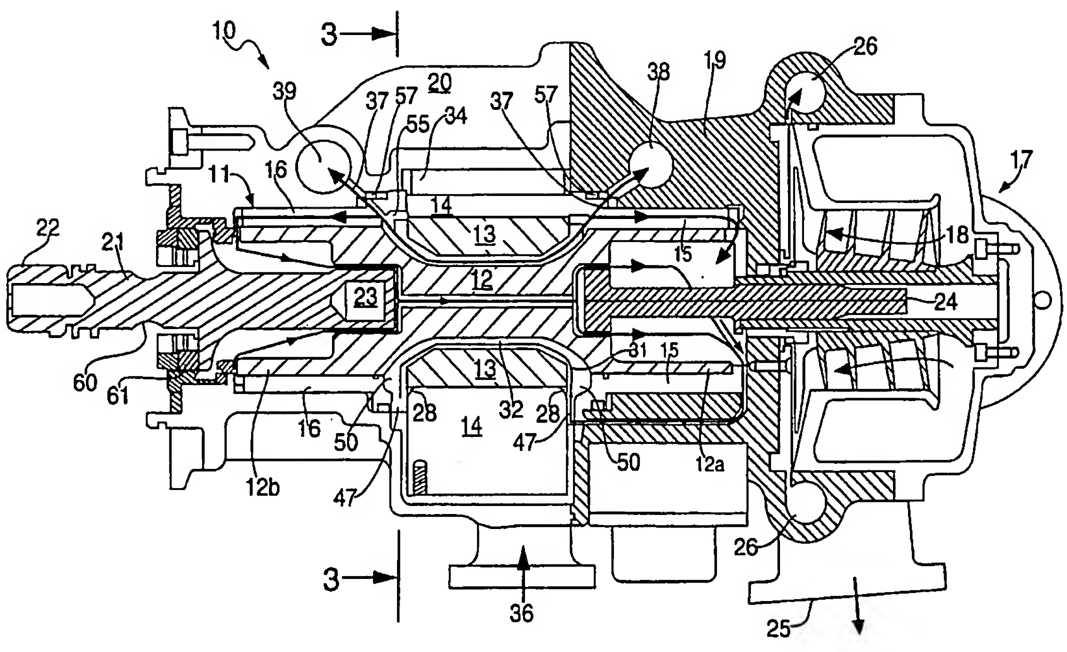
4,183,723	1/1980	Hansen et al.	417/204
4,222,712	9/1980	Huber et al.	417/204
4,354,809	10/1982	Sundberg	418/268
4,516,920	5/1985	Shibuya	418/268
4,913,636	4/1990	Niemiec	418/268
5,141,418	8/1992	Ohtaki et al.	418/30

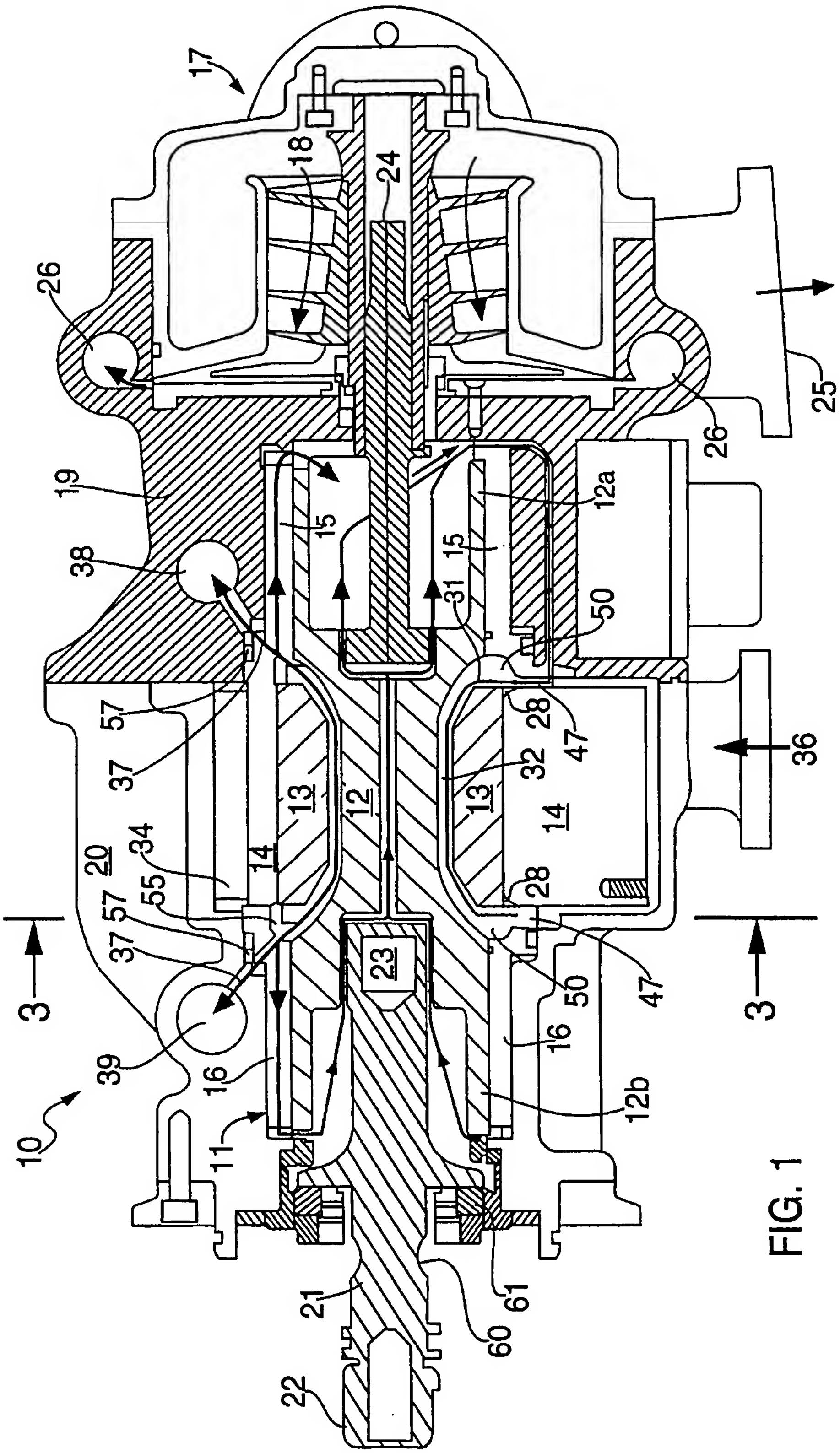
Primary Examiner—Charles Freay
Assistant Examiner—William Wicker
Attorney, Agent, or Firm—Howard S. Reiter

[57] **ABSTRACT**

A durable, single action, variable displacement vane pump capable of undervane pumping, components thereof, and pressure balancing method. The pump comprises a cylindrical barstock rotor member having large diameter journal ends and central vane slots uniformly spaced therearound. The vane slots are elongate and have a central vane-supporting portion of maximum depth surrounded at each end by extension portions having depths which decrease axially to the surface of rotor member. The varied rotor is rotatably supported within a unitary cam member having opposed faces and a circular bore therethrough forming a cam chamber having a continuous interior circular cam surface. The vane slot extensions in the rotor project outwardly beyond the cam chamber. An opposed pair of manifold bearings rotatably support the journal ends of the rotor and overlap the vane slot extensions to admit fluid to expanding vane bucket areas of the rotating vaned rotor and also into the vane slot extensions and undervane areas for pressure balancing purposes. Fluid passages and pressures within the pump are arranged to balance forces acting on various parts to reduce stress, improve sealing, and permit sharing of a fluid pressure source.

19 Claims, 8 Drawing Sheets




17
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12b
21
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61
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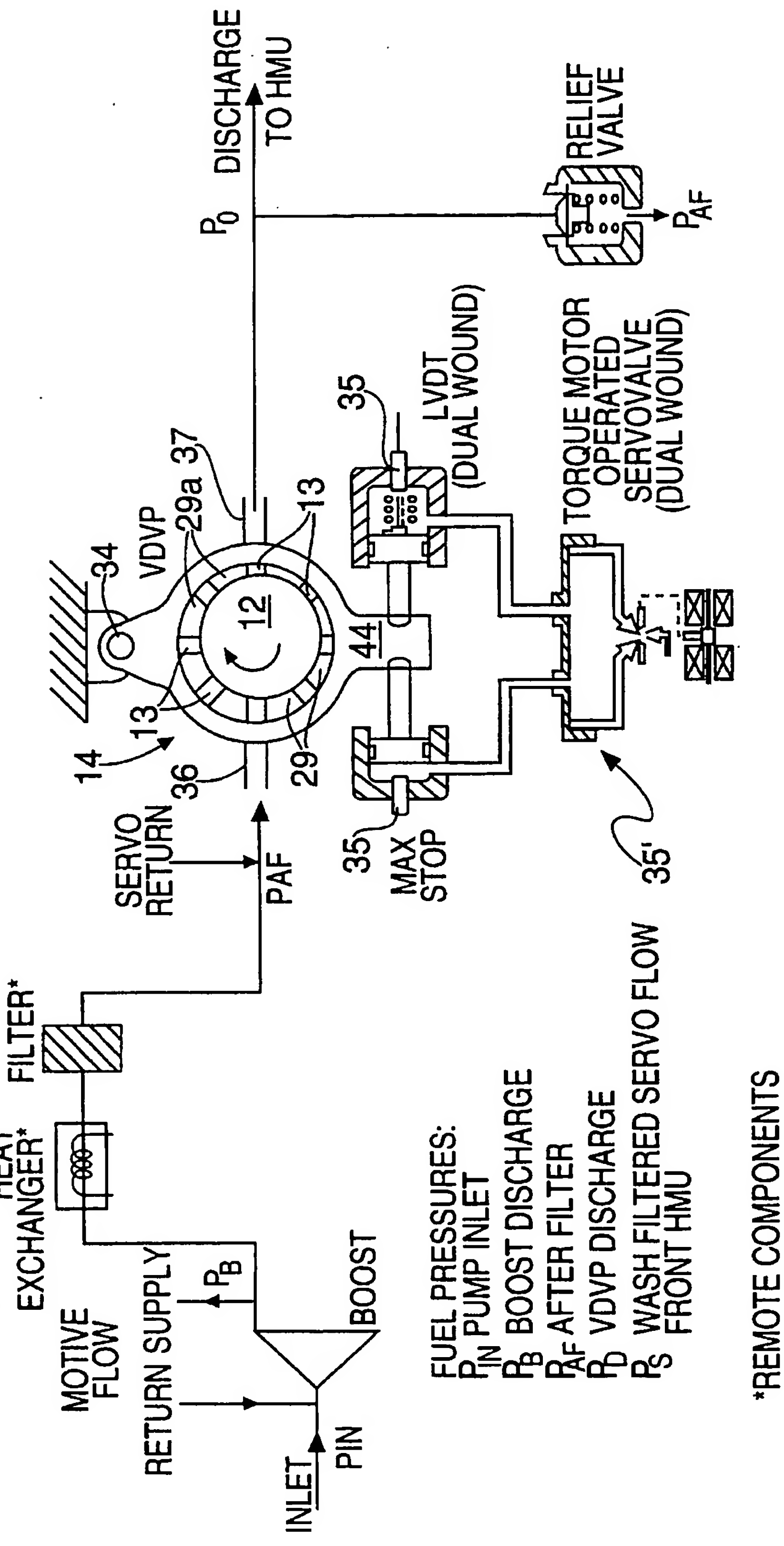
FIG. 1

FIG. 2



P_0 DISCHARGE
TO HMU
RELIEF VALVE
P_{AF}
VDVP
34
29a
37
13
12
44
14
36
29
35
LVDT (DUAL WOUND)
TORQUE MOTOR OPERATED SERVOVALVE (DUAL WOUND)
MAX STOP
35'
SERVO RETURN
PAF
FILTER*
HEAT EXCHANGER*
MOTIVE FLOW
RETURN SUPPLY
P_B
BOOST
INLET
PIN
FUEL PRESSURES:
P_{IN} PUMP INLET
P_B BOOST DISCHARGE
P_{AF} AFTER FILTER
P_D VDVP DISCHARGE
P_S WASH FILTERED SERVO FLOW FRONT HMU
*REMOTE COMPONENTS

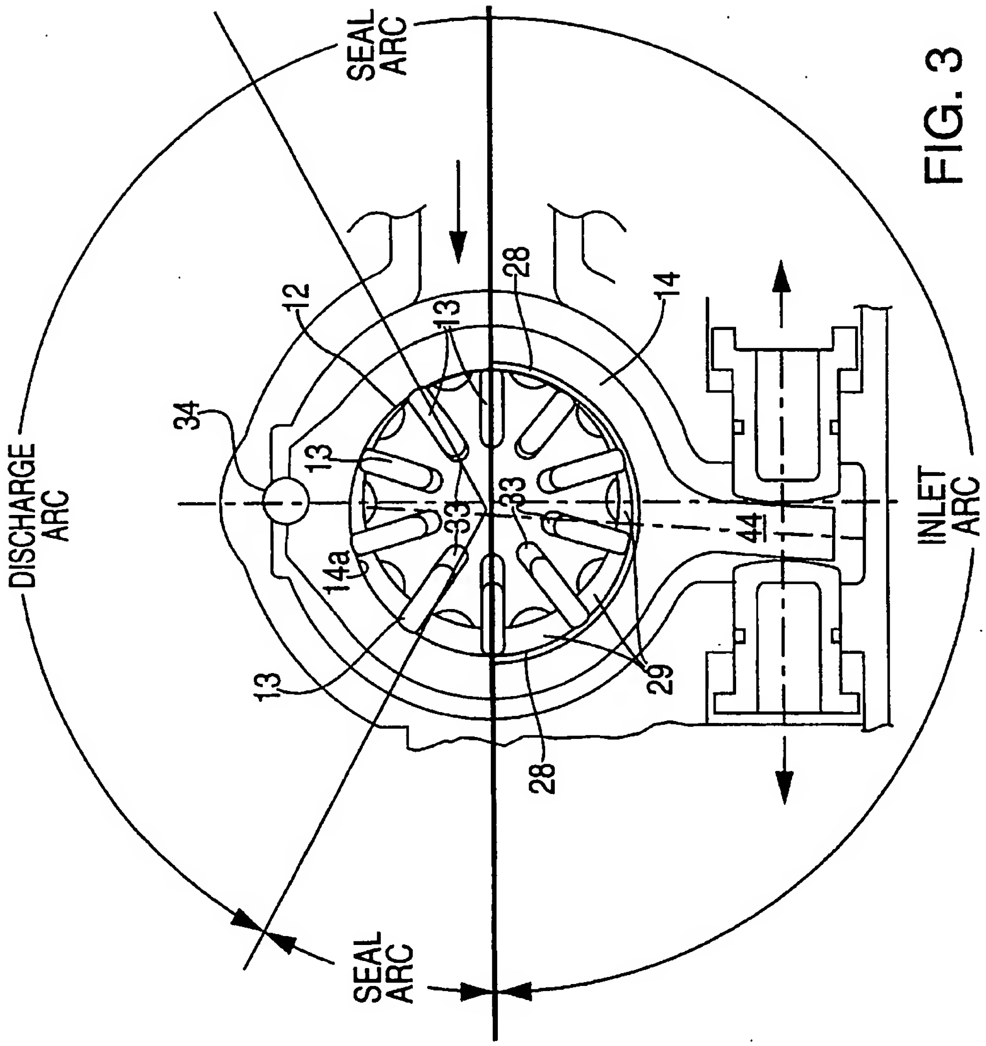

SEAL ARC
DISCHARGE ARC
INLET ARC
SEAL ARC
12
13
14
14a
28
29
33
34
44

FIG. 3

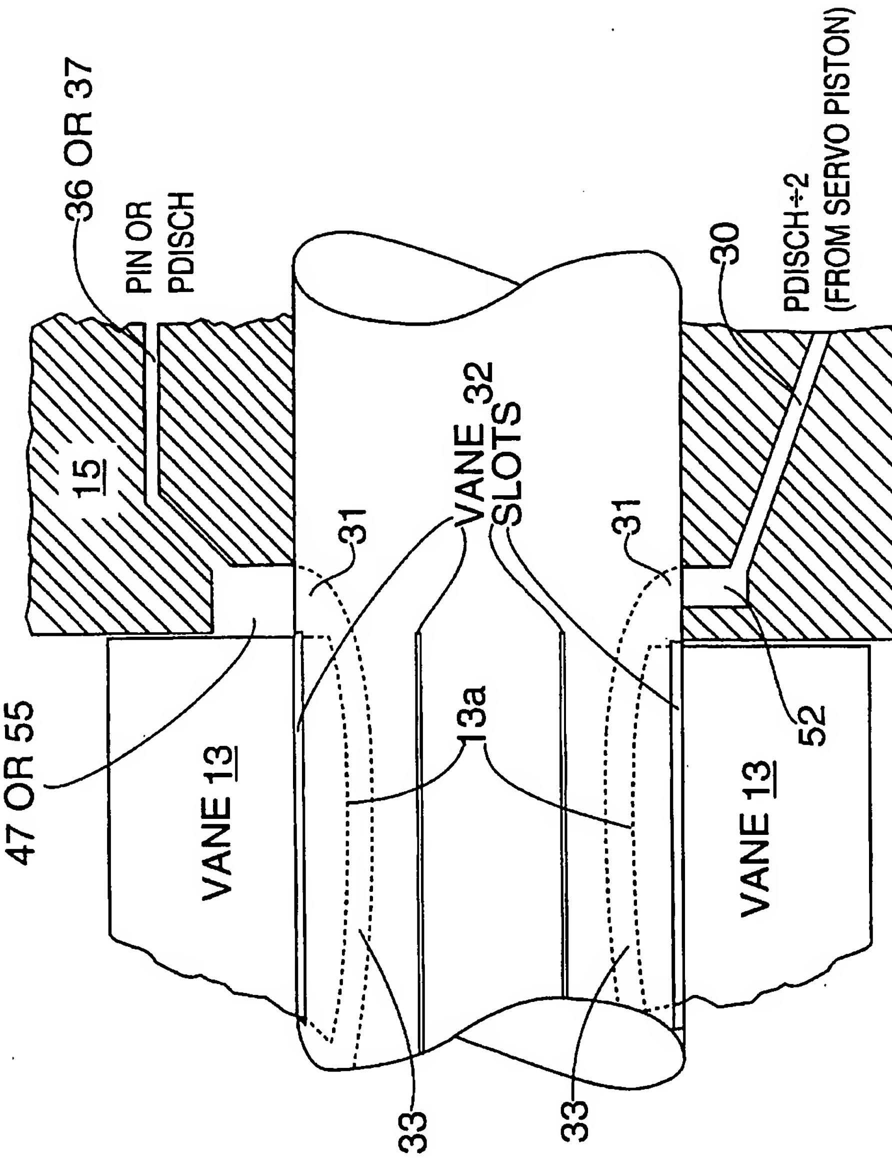

36 OR 37
PIN OR
PDISCH
15
VANE
SLOTS
32
31
31
30
PDISCH÷2
(FROM SERVO PISTON)
47 OR 55
VANE 13
13a
VANE 13
52
33
33

FIG. 4

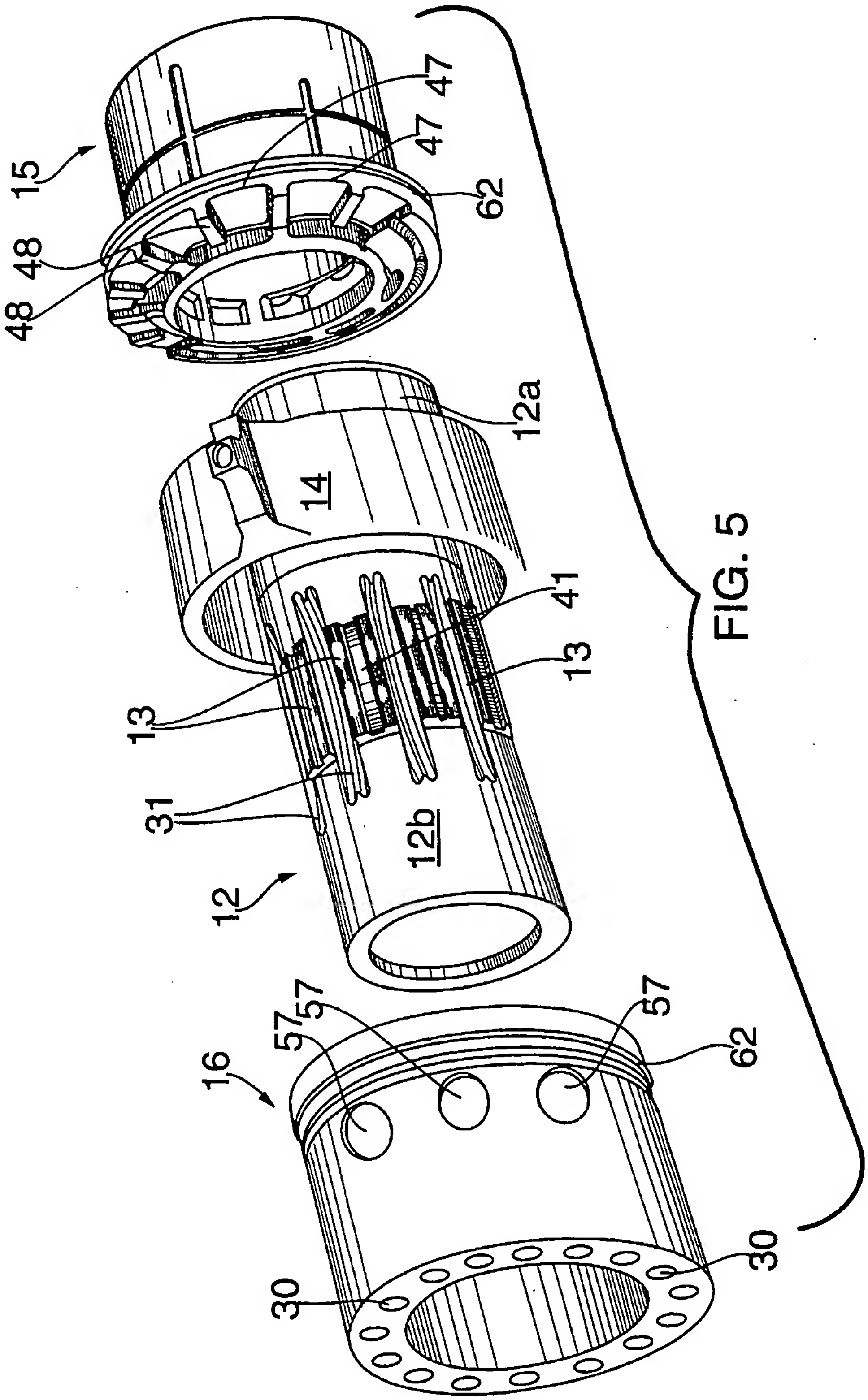


FIG. 5

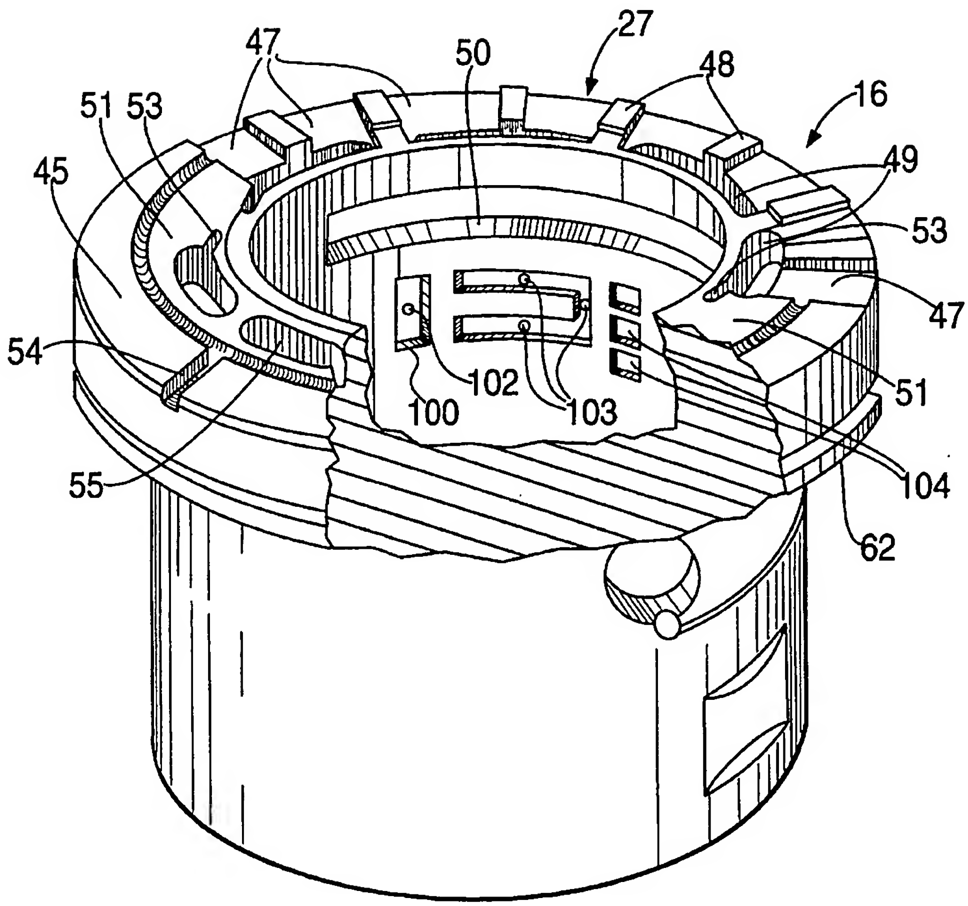


FIG. 6

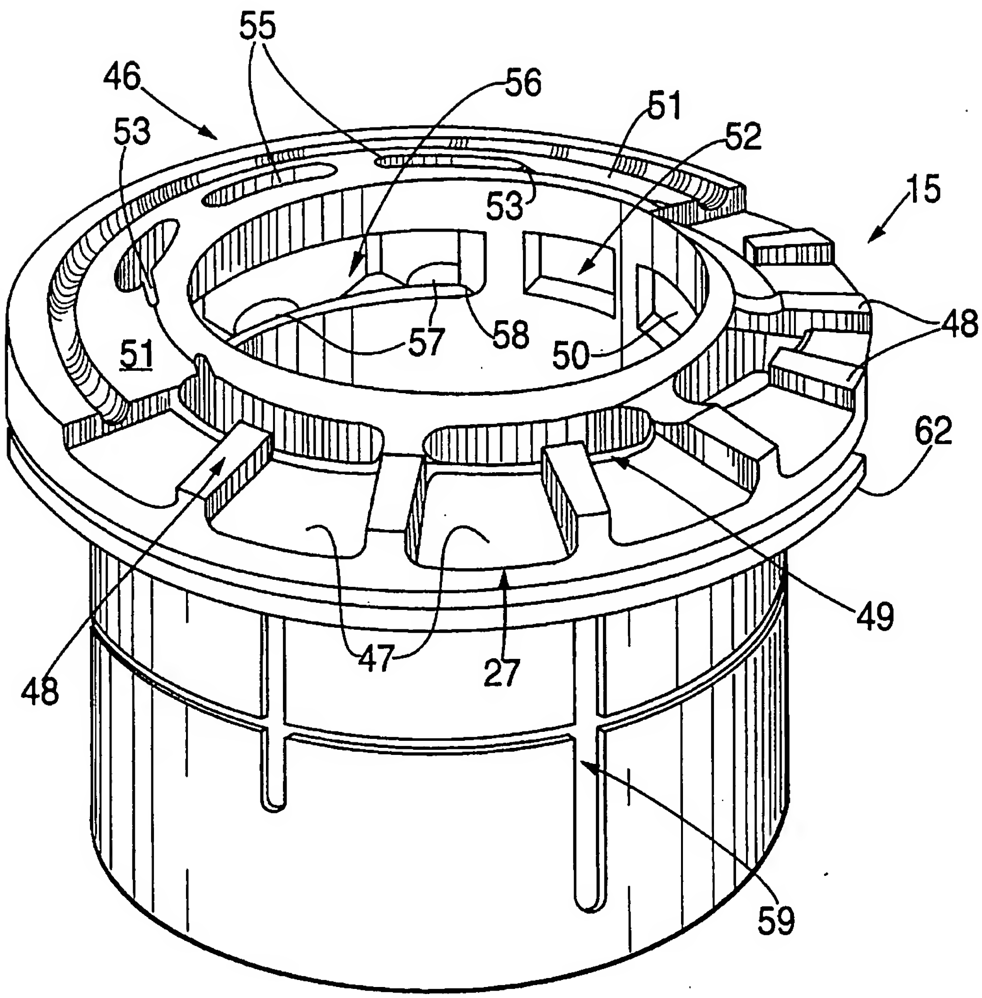
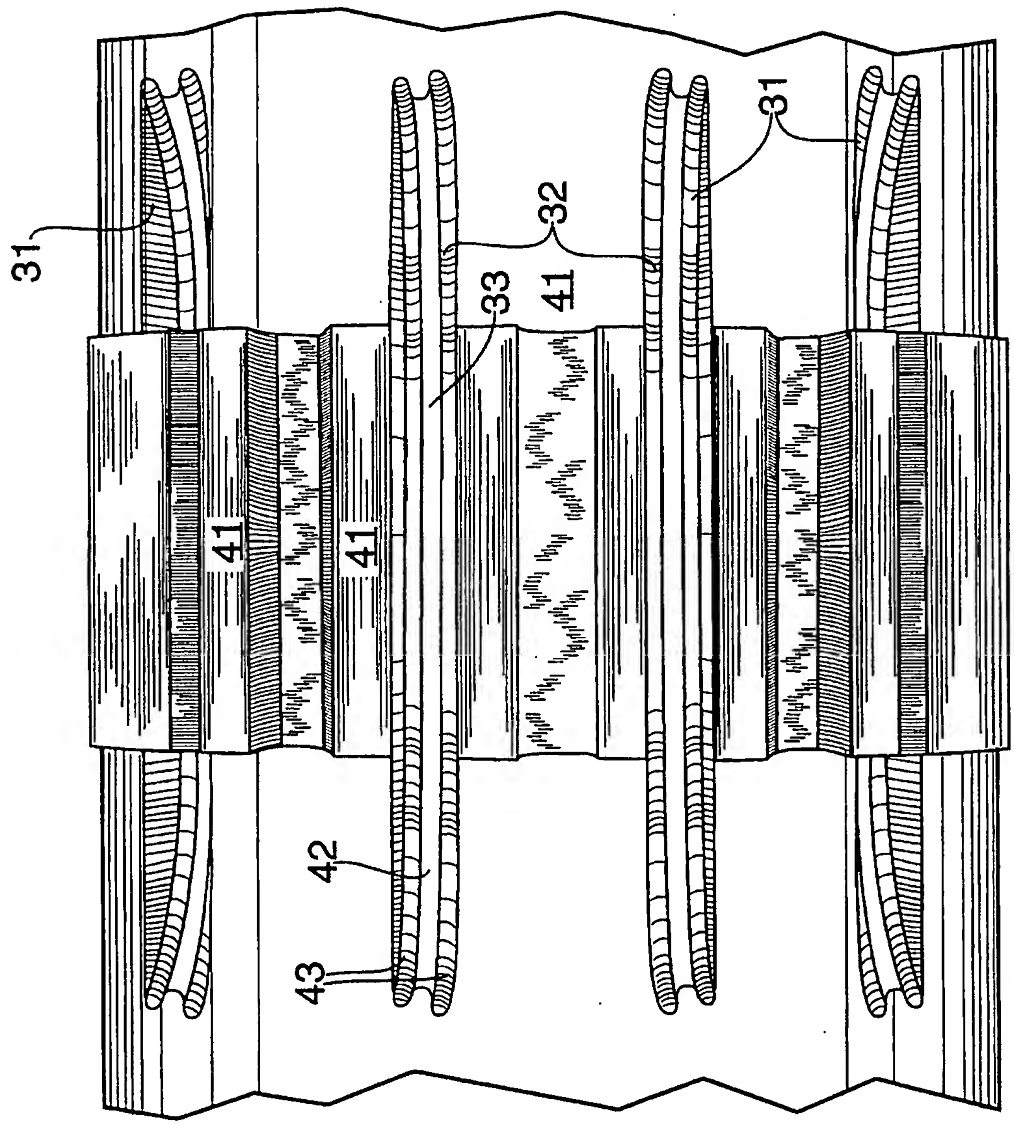


FIG. 7

FIG. 8



VARIABLE DISPLACEMENT VANE PUMP, COMPONENT PARTS AND METHOD

BACKGROUND OF THE INVENTION

1. Field of the Invention

The present invention relates to single acting, variable displacement fluid pressure vane pumps and motors, such as fuel and hydraulic control pumps and motors for aircraft use, component parts thereof and to a method for balancing fluid pressures.

Over the years, the standard of the commercial aviation gas turbine industry for main engine fuel pumps has been a single element, pressure-loaded, involute gear stage charged with a centrifugal boost stage. Such gear pumps are simple and extremely durable, although heavy and inefficient. However, such gear pumps are fixed displacement pumps which deliver uniform amounts of fluid, such as fuel, under all operating conditions. Certain operating conditions require different volumes of liquid, and it is desirable and/or necessary to vary the liquid supply, by means such as bypass systems which can cause overheating of the fuel or hydraulic fluid and which require heat transfer cooling components that add to the cost and the weight of the system.

2. State of the Art

Vane pumps and systems have been developed in order to overcome some of the deficiencies of gear pumps, and reference is made to the following U.S. Patents for their disclosures of several such pumps and systems: U.S. Pat. Nos. 4,247,263; 4,354,809; 4,529,361 and 4,711,619.

Vane pumps comprise a rotor element machined with slots supporting radially-movable vane elements, mounted within a cam member and manifold having fluid inlet and outlet ports in the cam surface through which the fluid is fed radially to the inlet areas or buckets of the rotor surface for compression and from the outlet areas or buckets of the rotor surface as pressurized fluid.

Vane pumps that are required to operate at high speeds and pressures preferably employ hydrostatically (pressure) balanced vanes for maintaining vane contact with the cam surface in seal arcs and for minimizing frictional wear. Such pumps may also include rounded vane tips to reduce vane-to-cam surface stresses. Examples of vane pumps having pressure-balanced vanes which are also adapted to provide undervane pumping, may be found in U.S. Pat. Nos. 3,711,227 and 4,354,809. The latter patent discloses a vane pump incorporating undervane pumping wherein the vanes are hydraulically balanced in not only the inlet and discharge areas but also in the seal arcs whereby the resultant pressure forces on a vane cannot displace it from engagement with a seal arc.

Variable displacement vane pumps are known which contain a swing cam element which is adjustable or pivotable, relative to the rotor element, in order to change the relative volumes of the inlet and outlet or discharge buckets and thereby vary the displacement capacity of the pump.

Among the disadvantages of known vane pumps are their lack of durability, susceptibility to wear, complexity of rotor and cam structures, necessity for end sealing plates to seal the ends of the rotor for the purpose of containing the pressurized fluid, and other essential elements which can provide vane pumps with variable metering properties not possessed by gear pumps but which detract from their durability or life span relative to the comparative durability

and life spans of gear pumps. In conventional vane pumps the rotor is splined upon and driven by a central drive shaft having small diameter journal ends/which are not strong enough to withstand the opposed inlet and outlet hydraulic pressure forces generated during normal operation. This problem is overcome by forming such pumps as double-acting pumps having opposed inlet arcs and opposed outlet or discharge arcs which balance the forces exerted upon the journal ends, as disclosed by the prior art such as U.S. Pat. Nos. 4,354,809 and 4,529,361, for example.

SUMMARY OF THE INVENTION

The present invention relates to novel single acting, variable displacement vane pumps, and components thereof, which have the durability, ruggedness and simplicity of conventional gear pumps, and the versatility and variable metering properties of vane pumps, while incorporating novel features and properties not heretofore possessed by prior known pumps of either type.

The novel pump of the present invention comprises a durable, substantially uniform diameter rotor member which may be machined from barstock, similar in manner and appearance to the main pumping gear of a gear pump. The rotor has large diameter journal ends at each side of a central vane section which includes a plurality of axially-elongated radial vane slots having central deeper well areas, slidably engaging a mating vane element. The rotor slots are such that the vanes may be significantly greater in thickness than is permitted in pumps constructed in accordance with the prior art. Axial grooves or depressions may be included in the surface of the rotor between the vane slots. These depressions provide increased volume, to reduce sudden pressure build-up which can occur when the enclosed volume between the vanes is reduced as it is during the pumping process. This can create an effect similar to "water hammer" in a residential plumbing system. An adjustable, narrow cam member having a continuous circular inner cam surface eccentrically surrounds and encloses the central vane section, and the cam surface is engaged by the outer surfaces of the vane elements during operation of the pump. The cam housing pivots a pin to provide the means for adjusting the operating "displacement" of the pump. Pressure forces within the cam are directed, through the porting structures of the bearings, so that the cam loads are centrally (i.e., symmetrically) located relative to the pin, thereby reducing the force needed to actuate the cam and reducing the stresses on the pin. This arrangement permits forces to be distributed so that the pin is maintained in compression, thereby simplifying alignment and assembly of the cam to the pin. The pin includes a crowned alignment feature which assures that the cam and the bearings will always be in close proximity. The journal ends of the rotor member are rotatably supported within opposed durable manifold bearings, which may be made for example from barstock material, and which have manifold faces which contact opposite faces of the cam member and overlap the outer ends of the elongated radial vane slots. Each manifold bearing has interior inlet and discharge passages communicating with the cam—contacting manifold faces. The latter comprise an inlet arc segment opening to the inlet passages of the bearing, and a smaller discharge arc segment opening to the discharge passages of the bearing, separated from each other by opposed small sealing arc segments. Rotation of the journals of the vaned rotor member within the manifold bearings and of the central vane section within the cam member causes fluid such as liquid fuel to be admitted axially through the inlet

arc segments of the bearings into the cam chamber and into expanding inlet bucket chambers between the vanes, and also through the inlet manifold passages and the vane slot extensions to under-vane chambers. Continued rotation of the rotor member through a sealing arc segment into a discharge arc segment changes the pressure acting upon the leading face of each vane from inlet pressure to increasing discharge pressure as the volume of each bucket chamber is gradually compressed at the discharge side or arc of the eccentric cam chamber. The pressurized fuel escapes into the discharge ports of each manifold bearing, through the discharge passages, and is channelled to its desired destination.

According to the present invention, the pressures acting upon the vanes are balanced so that the vanes are lightly loaded or "floated" throughout the operation of the present pumps. This reduces wear on the vanes, permits the use of thicker, more durable vanes and, most importantly, provides elasto-hydrodynamic lubrication of the interface of the vane tips and the continuous cam surface. Such balancing is made possible by venting the undervane slot areas to an intermediate fluid pressure in the seal arc segments of the manifold bearings whereby, as each vane is rotated from the low pressure inlet segment to the high pressure discharge segment, and vice versa, the pressure in the undervane slot areas is automatically regulated to an intermediate pressure at the seal arc segments, whereby the undervane and overvane pressures are balanced which prevents the vane elements from being either urged against the cam surface with excessive force or from losing contact with the cam surface. The intermediate pressure at the seal arc segments is derived from the servo piston pressure which is used to move the cam.

The regulation of the undervane pressure permits the use of thicker, more durable vanes by eliminating the unbalanced pressures which are found in the prior art. In the prior art, vanes are made thin to limit the loading of the vane against the cam, because relatively high discharge pressure produces the force that urges the vane tip against the cam, while relatively low inlet pressure acts to relieve the interface pressure between the tip and the cam. The small area of the thin vane allows tolerable loads at the vane tip but often requires dense brittle alloys and results in fragile vanes. Within the inlet arcs of the present invention the undervane areas are subjected to inlet pressure as are the overvane areas. Within the outlet arcs of the pump, the undervane areas are subjected to outlet pressure as are the overvane areas. Within the seal arcs of the pump, the undervane areas are subjected to a pressure that is midway between inlet and discharge pressure, to compensate for the overvane areas which are also subjected half to inlet and half to discharge. More importantly, the regulation of the undervane pressure and "floating" of the vanes causes the outer surfaces of the vanes to float over the continuous cam surface which is lubricated by the fluid being pumped, whereby metal-to-metal contact and wear are virtually eliminated. This overcomes the need for hard, brittle, wear-resistant, heavy metals, such as tungsten carbide, for the vanes and/or for the cam surface and permits the use of softer, more ductile, lightweight metals, particularly if the outer vane tips are radiused or rounded and a wear resistant coating, such as of titanium nitride, is applied to the outer rounded vane tip surfaces and to the cam surface.

The structural features of the journal bearing include a "hybrid" bearing pad which is supplied with discharge pressure from the pump. The discharge pressure provides a high load level bias which increases the load carrying capability of the bearing. The pad is configured with a single, axial pressure-fed groove, which provides lubricant and a pressure bias on the incoming rotor direction. The pad also includes a "U" shaped groove with the legs of the "U" positioned transverse to the axis of the journal bearing and the bottom of the "U" being located on the outgoing rotor direction. These legs and bottom of the "U" shaped groove are supplied with high pressure lubricating fluid to provide a desired pressure bias. The journal bearing structure further includes a larger diameter, eccentrically located flange on the face, which contacts the cam to assure that the bearings have sufficient load to maintain contact with the cam. The surface of the flange adjacent to the cam includes relief grooves to minimize the amount of face area which is subjected to discharge pressure induced outward load, from the cam. The surface of the flange most distant from the cam is loaded in its entirety with discharge pressure to assure that the net load acts against the cam. The eccentric favors increased area in the discharge pressure arc to assure that the loading is always against the cam. The top inner diameter of the bearing, for a distance around the sides slightly away from the hybrid pressure pad, contains labyrinth seal grooves for the purpose of limiting the amount of parasitic bearing flow.

The bearing seal-arc ports are located entirely above the horizontal centerline of the rotor with the bottom of these ports not being positioned below the centerline. In this manner, the ports will not be located in a region where the volume of the vane buckets is increasing, because expansion of the bucket volume in the seal area region tends to produce destructive cavitation. The ports, being above the centerline will permit only slight compression of the vane buckets, thereby avoiding the potential for cavitation.

The novel vane pumps of the present invention also provide substantial undervane pumping of the fluid from the undervane slot areas by piston action as the vanes are depressed into the slots at the discharge side of the cam chamber. Such undervane pumping can contribute up to 40% or more of the total fluid displacement.

DESCRIPTION OF THE DRAWINGS

FIG. 1 is a schematic cross-sectional view of a fuel pump assembly according to one embodiment of the present invention, illustrating fluid flow paths therethrough;

FIG. 2 is a schematic diagram of the fuel pumping system through the assembly of FIG. 1, including an adjustment system for the cam member to vary the fuel displacement volume;

FIG. 3 is a schematic cross-sectional view of the single acting vane stage of FIG. 1 taken along the line 3—3 thereof;

FIG. 4 is a simplified schematic depiction of the supply or discharge of fluid to or from the undervane slot areas in the areas of the inlet and discharge arcs respectfully, and of the porting of the undervane slot areas to an intermediate, balancing pressure in the areas of the seal arcs of the cam chamber;

FIG. 5 is a perspective view of a single acting vane stage comprising a substantially uniform-diameter rotor member, containing vanes, a cam member and manifold bearing members according to the present invention, the members being shown in disassembled configuration for purposes of illustration;

FIG. 6 is a partially cut-away perspective view of the pressure pad of the manifold bearing members of FIG. 4 viewed from one end thereof;

FIG. 7 is a perspective view of the manifold bearing members of FIG. 6, viewed from the opposite end thereof; and

FIG. 8 is an enlarged perspective view of the central slotted area of the rotor member of FIG. 5, with the vane elements removed to illustrate the novel configuration of the vane slots therein.

DETAILED DESCRIPTION

Referring to FIG. 1, the fuel pump assembly **10** thereof comprises a variable displacement single acting vane pump **11** having a rugged barstock rotor member **12** having a plurality of vane elements **13** radially-supported within axially-elongated, concave vane slots **32** disposed around the central area of the rotor member **12**. The outer tips of the vane elements **13** preferably are rounded to reduce their areas of contact with the interior continuous surface **14a** (FIG. 3) of an adjustable cam member **14**, and a pair of manifold bearing blocks or members **15** and **16** rotatably support the large diameter journal ends **12*a*** and **12*b*** of the rotor member **12** and provide axial sealing of the pressurized chamber. In this regard, the blocks **15** and **16** serve the function of the "side" or "end" plates of a conventional vane pump.

The vane pump **11** is fed with fluid from a centrifugal boost stage **17** comprising an axial inducer and radial impeller **18** and associated collector and diffuser means **26** mounted within a housing section **19** connected to a housing section **20** mountable on a main engine gearbox.

Power is extracted in conventional manner from an engine through a main drive shaft **21** which includes an oil-lubricated main drive spline **22**, a fuel-lubricated internal drive spline **23**, a shear section **60** and a main shaft seal **61**. A second shaft **24** drives the boost stage **17** from a common spline with the main shaft **21**.

The pump is mounted to the main engine gearbox, and ports are provided to passages through the housing section **19** for an outlet **25** from the boost stage **17** through diffuser means **26** to an external heat exchanger and filter (FIG. 2) and back into inlet passage **36** (FIG. 2) to the inlet arc section **27** of the manifold bearings **15** and **16** for axial introduction of the fuel, under inlet pressure, past the hemispherical bevels or undercut slots **28** on the opposed faces of the cam member **14** in the area of the inlet arc of the cam chamber and into the expanding fuel inlet buckets **29** formed between adjacent vane elements **13** within the inlet arc section of the cam member **14**, as shown in FIG. 3.

Rotation of the rotor **12** and vanes **13** within the cam member **14** causes the inlet buckets **29** to move into a seal arc area where they become isolated from the inlet arc sections **27** of the manifold bearings **15** and **16** and begin to become compressed due to the non-concentric axial position of the rotor member **12** within the cam chamber, as shown in FIG. 3. Within the seal arc zones, which are transition zones between the lower-pressurized inlet pressure zone and the increased discharge pressure zone, each vane experiences a different overvane pressure on each side of it, which normally can cause intermediate overvane forces. However, as illustrated by FIG. 4, the present pumps provide special pressure relief passages **30** to a source of fluid at intermediate pressure in the seal arc areas whereby fuel is supplied at intermediate pressure through axial passages **30** in the manifold bearings **15** and **16** (FIG. 5) to the extremities **31** of the vane slots **32**, beyond the vane elements **13**, to produce an intermediate fluid pressure in the undervane slot areas **33** which balances the overvane fluid pressures and reduces the stresses or forces exerted by the vane tip surfaces against the continuous cam surface **14a** in the area of the sealing arc zones. As can be seen from FIGS. **3** and **4**, the undervane areas **33** are biased directly to inlet pressure, through slot extensions **31** and bearing ports and passages when the vane is in the inlet arc, and to discharge pressure when the vane is rotated to the discharge arc zone. In this manner, the vane loading in the inlet, seal, and discharge arc zones is held to very tolerable levels since the vane loads are achieved primarily through a combination of balanced pressure forces an low dynamic forces.

FIG. **2** is a simplified depiction of a cam member mechanism adjustable between minimum and maximum displacement flow positions. The cam **14** pivots on a pin **34** supported within housing section **20** at the top of the pump structure member. The pump is at maximum displacement when the cam **14** is positioned so that the vane buckets experience maximum contraction in the discharge arc zone. Likewise, minimum flow occurs when the cam **14** and the rotor **12** are almost concentric. Mechanical stops **35** are designed into a piston adjustment system **35'** to limit cam displacement, generally, for the purpose of assuring that the cam will not contact the rotor surface (exceeds max displacement). These stops include shims for final production calibration. The piston adjustment system **35'** is supplied with fluid at a predetermined pressure selected to be "intermediate" or "half-way" between the inlet and discharge pressures of the pump. This arrangement permits the use of a common source of fluid pressure (not shown) for both the adjustment system **35'** and the axial relief pressure passages **30** and associated sealing arc ports **52** shown in FIG. **4** and described elsewhere herein.

As illustrated by FIGS. **1** and **2**, the fuel exits the booster stage **17** of the pump through an external flanged outlet **25** and a collector/diffuser means **26** from the axial inducer/impeller **18** at the front of the boost stage **17**. The axial inducer imparts sufficient pressure rise to the fluid to eliminate poor quality effects associated with line losses or fuel boiling and assures that the main impeller, downstream from the inducer, will be handling non-vaporous liquid. Angled slots in the impeller hub allow some of the flow to move from the front to the back side of the impeller. Hence fuel passes radially outward through the vaned passages on both sides of the impeller, subsequently to be collected and diffused. As shown in FIG. **2**, the fuel exits the booster stage **17** through outlet **25** to pass through the external engine heat exchanger and filter, subsequently, to return, via an inlet passage **36** in housing section **20**, to the main vane stage. Fuel enters around the main vane stage cam **14** in the inlet arc zone **27** and is admitted, axially, to the expanding inlet vane buckets **29** through an undercut slot **28** on each cam face from face recesses in each of the bearings **15** and **16** and on both sides of the cam **14**. Each vane bucket **29** then carries the fuel circumferentially into the discharge arc where contracting discharge bucket **29*a*** squeeze the fuel axially outward into discharge ports **55** (FIG. **7**) cut into the faces of the bearings **15** and **16** in the discharge arc zone, subsequently to be discharged to the engine through cored passages **38** and **39** in the housing sections **19** and **20**. FIG. **1** provides a depiction of the flow path through the system.

Certain prior art vane pumps were designed to perform in the absence of a filter and therefor intimate working parts, including cams, vanes and sideplates, were fabricated from tungsten carbide, a very tough, dense, brittle material. The high density of the vanes resulted in high centrifugal loading which, when combined with the substantial pressure loads

under the vanes in the inlet and sealing arcs, demanded that the vanes be very narrow in order to minimize vane loading/wear at the interface with the cam. Through the incorporation of filtered fuel as the means for contamination resistance, and the use of pressure balancing as the means for moderating the forces acting on the vanes, a lower density, more ductile high vanadium-content tool steel alloy material is used according to the present invention, thereby assuring a far less fragile pumping vane and cam.

The novel design of the present pumps enables the use of thicker vanes which obviously have lower bending stress and greater column stiffness. A less obvious but very important corollary to the effect of thicker vanes is that the vane tip radius can be much greater (a factor of five), thereby permitting configuration of the vane tip as a continuous, smooth surface for the enhancement of vane tip lubrication at the interface with the continuous cam surface **14**a.

In addition to balancing the undervane and overvane loads on the vane elements **13**, the undervane access and capacity through the downwardly-tapered vane slot extensions **31** increases the volumetric capacity of the pump by enabling the introduction and discharge of undervane fluids to and from undervane areas **33**. As the vane passes through the inlet arc, the cavity **33** under the vane **13** is filled with fuel as the vane expands out of the vane slot **32**. As the vane passes through the discharge arc, the downward movement of each vane **13** into its slot **32** forces that fluid out of each undervane cavity **33**, resulting in a pumping action which greatly increases the capacity of the pump. The present pumps have thick vanes and can extract almost 40% of capacity from undervane pumping. The vane elements **13** fit snugly within the vane slots **32** and function like pistons as they are depressed into the arcuate slots **32** during movement of the rotor through the discharge arc, whereby fluid is expelled axially from the undervane areas **33** outwardly in both directions through the slot extensions **31**, discharge ports **37** and cored passages **38** and **39**. The bulk of the pressurized discharge fluid or fuel is expelled from the bucket areas **29**a, between vane elements **13**, but the undervane volume from cavities **33** can equal as much as about 40% of the total discharge volume. Referring to FIGS. **5** to **8** of the present drawings, these illustrate in greater detail the rugged, robust barstock rotor member **12** (FIGS. **5** and **8**), vane elements **13** (FIG. **5**), cam member **14** (FIG. **5**) and manifold bearings **15** and **16** (FIGS. **5** to **7**).

The rotor member **12** has an appearance and shape similar to a conventional heavyweight gear shaft in that it has a substantially uniform thick diameter throughout, and a central vane area **40** comprising optional spaced radial teeth **41** which provide additional support for the vane elements **13** in areas above the vane slots **32** cut into the rotor cylinder. Between every other pair of said teeth **41** a contoured arcuate vane slot **32** is machined radially into the rotor to receive a relatively thick vane element **13** having an axial length similar to the length of the teeth **41** and of the central vane area **40** so that each vane **13** occupies only the central, deep area of each arcuate or contoured slot **32**, and the outwardly-tapered extremities **31** of each slot **32** are open beneath the adjacent undersurface areas of the manifold bearings **15** and **16**. Moreover the contoured seat areas **42** of each slot **32** are raised stop areas between deeper well or floor areas **43** to provide undervane areas or cavities **33** even if the contoured undersurface **13**a of the vanes **13** (shown in FIG. 4) is depressed into contact with the raised seat recesses **42**.

As can be noted, the undervane regions and cavities **33** are open at slot areas **31** directly to inlet pressure when each vane element **13** is in the inlet arc, and directly to discharge pressure when each vane element **13** is located in the discharge arc region. In this manner, the vane loading in the inlet and seal arcs is held to very tolerable levels since the vane loads are achieved primarily through dynamic forces. Within the seal arcs, the transition region between inlet and discharge (and vice-versa), each vane **13** normally would experience a different pressure on each side of it, resulting in intermediate overvane forces which must be counteracted. However, sealing arc ports **52** are provided in the inner diameter walls of the bearings **15** and **16**, between the inlet and discharge arc zones, which communicate through axial relief pressure passages **30** in the bearing walls with a fluid source at an intermediate pressure level, approximately halfway between inlet and discharge pressures, as shown by FIG. 4.

Prior-known vane pumps utilized discharge pressure under the vanes to assure that the vanes properly tracked the cam surface in all areas of operation. That approach was to assure that the vane trajectory followed the cam contour. The resulting high forces, especially in the inlet arc, yielded a propensity for wear at the tip of the vanes. The present invention utilizes the resident pressure in the inlet and discharge arc areas or zones and a regulated intermediate level of pressure in the sealing arc areas or zones to provide a balancing pressure under the vanes. This assures that each vane element **13** will always track the continuous cam surface **14**a on an elasto-hydrodynamic film, thereby assuring long life at the vane tip wearing surfaces. Vane speeds (pump RPM) are held at levels which provide sufficient residence time to assure that the vane trajectory will properly track the cam surface.

In the inlet and discharge arc, shown in FIG. **3**, the overvane and undervane pressures are equal. In the seal arc where the overvane sees inlet pressure on ½ of its tip and discharge pressure on the other ½ half of its tip, the undervane cavity **33** is ported to a servo piston chamber which is at approximately ½ discharge pressure. Thus the vanes **13** are pressure balanced or floated throughout the entire revolution, thereby reducing centrifugal stress forces and wear at the interface between the rounded vane element surfaces and the continuous surface **14**a of each cam element **14**, enabling the use of thicker, stronger vane elements and producing elasto-hydrodynamic lubrication at said interface.

The rugged, one-piece cam element **14** of FIGS. **2**, **3** and **5** is machined from a solid ingot, such as of high vanadium-content tool steel alloy. The cam element is banjo-shaped, having a circular axial bore or cam chamber in the middle for containment of the central vane area **40** of the vaned rotor section, a pivot shaft or pin **34** at the top which provides the fulcrum for the variability feature, and an extension **44** at the bottom which provides a lever for exerting adjustment force to vary the displacement. A generous chamfer bevel or slot **28** exists within the inlet arc on both cam faces to facilitate the introduction of the fuel into the expanding vane buckets **29**.

The pivot pin or shaft **34** is a simple cylinder, made of any suitable high strength alloy such as high vanadium content tool steel alloy coated with titanium nitride, which engages a cam pivot notch and a seat in the housing section **20**.

An important feature of the present cam elements **14** is the continuous smooth cam surface **14**a, shown in FIG. **3**, which is made possible by the axial fuel delivery and discharge means of the present pump assemblies. Prior-known variable displacement pumps contain interruptions in the cam

surface, such as radial inlet and discharge ports or a variable displacement parting line between cam sections which, however refined in edge treatment, are bound to cause irregularities in the operation of the vanes. In the case of two-piece vanes, necessitated by brittle material, special precautions had to be taken to assure that the vanes do not tilt into the openings, thereby causing destructive wear. The present pumps utilize an unbroken continuous cam surface **14***a* which provides uniform support of the vane elements **13** throughout their travel. This, coupled with the balancing of the undervane and overvane pressures and the elastohydrodynamic lubrication of the vane/cam interface, substantially reduces wear and increases the lifetime of the present pumps and components.

The present rotors **12**, shown in FIGS. **5** and **8**, differ substantially from prior known vane rotors since the latter have straight line, flat-bottom vane slots, parallel to the rotor axis, extending through sideplates, and require sideplates with undervane communication grooves and other features which necessitate the use of small-diameter journal shafts. Such shafts cannot withstand the opposed inlet and outlet forces of a single action pump and necessitate the incorporation of two opposed inlet and outlet stages for double action balance. The journal ends **12***a* and **12***b* of the present rotors are hefty, large diameter journals. Furthermore, the massive characteristic of the rotor **12** eliminates the structural weakness associated with vane slots being too close to the internal drive spline in prior known pumps. The strength of the rotor element **12** is complimented by the hefty nature of the identical manifold bearings **15** and **16** which rotatably receive and support the journal ends **12***a* and **12***b* of the rotor **12**.

As shown most clearly in FIGS. **6** and **7**, the manifold bearings **15** and **16**, are unitary machined elements incorporating the functions of a journal bearing, a face bearing and a sideplate. The bearings are designed for rugged, infinite life operation. The bearing material can be ductile leaded bronze alloy or a suitable equivalent. The bearing faces and inner diameter surfaces are treated with indium plating and dry film lubricants.

Each bearing face, which contacts a face of the cam member **14**, comprises an inlet arc section **27**, comprising about one-half of each face, an outlet or discharge arc section **45**, comprising a wide angle of less than 180 degrees and transition seal arc areas between the inlet arc and discharge arc section, comprising angles such that the sum of the discharge arc and the two seal arcs is 180 degrees.

Referring to FIGS. **6** and **7**, the bearing faces are machined or sculpted to provide an inlet half section **27** and a seal/discharge half section **46**. The inlet half section **27**, or 180° section, comprises radial face inlet recesses **47**, cut between stand-off radial face portions **48**, providing inlet recesses to inlet ports **49** opening into a arcuate common chamber **50** beneath the face of the inlet arc surface **27**, which opens to the inner-diameter surface of the bearings **15** and **16**. The stand-off radial face portions **48** of each bearing contact a face of the cam member **14**, as does the face of the seal/discharge half **46**, to assure uniform bearing strength for the loads associated with interaction with the cam member **14**.

Each bearing **15** and **16** has a face portion of increased diameter, compared to the remainder of the bearing, thereby providing a flange or shoulder **62** against which a spring-loading means can be biased to pressure-load the bearing faces against the opposed cam faces with sufficient force to prevent leakage of the pressurized fuel from the cam chamber.

As can be seen from the fuel flow illustration in FIG. **1**, the outer extremities or extensions **31** of the vane slots **32** extend beyond the cam member **14**, at each side thereof, and underlie the inner diameter surface of a bearing **15** or **16** so as to open the undervane areas **33** of the vane slots **32** to the inlet chamber **50** at the inlet side of the bearings **15** and **16**. Also, the recesses **47** of each bearing face communicate with an undercut slot **28** on an opposed face of the cam member **14**, and with an inlet passage **36**, to admit inlet fuel into the inlet buckets **29** or overvane areas, as illustrated by FIG. **4**.

Rotation of the rotor-vane pump moves each expanding inlet bucket **29** into axial opposition to the seal/discharge half **46** of the bearing faces where the overvane bucket areas move past the open inlet recesses **47** and over the closed seal arc face **51** which isolates the bucket areas from the inlet conduits but opens the undervane areas to an intermediate pressure fluid supply through the seal arc port **52** which communicates with the vane slot extensions **31** at the inside surface of each bearing **15** and **16**. Ports **52** open to isolated axial passages **30** (FIGS. **4** and **5**) within the bearings which communicate with a source of fluid at regulated pressure, intermediate the inlet and discharge pressures. However, eyelet cuts **53** are placed in the sealing arc face **51** to assure that the vane buckets within the sealing arcs cannot undergo unvented compression. This assures that the undervane areas **33** of the vane slots **32** are held within pressure limits during the period of time that the vane buckets pass through the intermediate regions between the inlet pressure and the discharge pressure arcs.

Continued movement of the vane buckets over the face **54** of the discharge arc section **45**, shown between broken lines in FIG. 7, opens the compressed buckets **29***a* to discharge ports **55** in face **54** as the buckets undergo compression due to the eccentric, non-concentric axial position of the cam member relative to the rotor/vane pump enclosed within the cam member **14**, as illustrated by FIG. 3. The discharge ports **55** are inlets to a common internal discharge chamber **56** having discharge outlet ports **57** in the outer diameter wall of the bearings **15** and **16** and having a common vane slot discharge port **58** in the inner diameter wall of the bearings to admit undervane pumping fluid discharge from the undervane areas **33** through the vane slot extensions **31**, as shown in FIGS. **1**, **5** and **7**. As illustrated by FIGS. **1** and **7**, the outer diameter discharge outlet ports **57** open radially outwardly to discharge passages **37** and conduits **38** and **39** in the housing to deliver the fluid or fuel at elevated discharge pressures to an engine, hydraulic system or other desired destination. The discharge ports **55** in face **54** are open axially to the contracting vane buckets **29***a* during their compression to admit the vane bucket volumes of the pressurized fluid, while the inner diameter port **58** is open to the vane slot extensions **31** to receive the fluid which is pumped from the undervane areas **33** (FIG. 3). This may represent up to about 40% of the total amount of fluid being pumped. Fluid is pumped from the undervane areas in this manner as the vane elements **13** are depressed into their slots **32** to compress and displace the undervane fluid axially in both directions from the undervane areas **33**, through the slot extensions **31**, and into the inner diameter bearing ports **58** to chamber **56** and outer diameter outlet ports **57**.

In summary, fuel enters the present pump assemblies **10** through an external inlet flange and a cored passage which leads to the axial inducer **18** at the front of the boost stage **17**. The axial inducer imparts sufficient pressure rise to the fluid to eliminate poor quality effects associated with line losses or fuel boiling and assures that the main impeller, downstream from the inducer, will be handling non-vapor-

ous liquid. Angled slots in the impeller hub allow some of the flow to move from the front to the back side of the impeller. Hence, fuel passes radially outward through the vaned passages **26** on both sides of the impeller, subsequently to be collected and diffused. The fuel leaves the pumping system through outlet **25** to pass through the engine heat exchanger and filter, subsequently to return, via a cored passage **36**, to the main vane stage. Fuel enters a plenum around the main vane stage cam and is admitted, axially, to the expanding inlet vane buckets **29** through an undercut slot **28** on both side faces of the cam **14**. Each vane bucket **29** then carries the fuel circumferentially into the discharge arc where the contracting bucket **29a** squeezes the fuel axially outward into ports **55** cut into the face of the manifold bearings **15** and **16**. The overvane bucket fuel is then discharged through chamber **56** and the bearing ports **57** into a port **37** between the bearing **15**, **16** and the housing **19**, **20** subsequently to be discharged to the engine through cored passages **38**, **39** in the housing. The undervane fuel is discharged through the vane slot extensions **31** into the discharge chamber **56** through the inner diameter port **58** to contribute up to about 40% of the total fuel pumped through the outer diameter ports **57**.

The manifold bearings **15** and **16** receive lubricant and cooling flow through two sources. The high pressure discharge arc **45** of the vane pump provides a source of pressure to force fuel axially through the diametral clearance between rotor journals **12a** and **12b** and bearings **15** and **16**. This flow is managed through careful clearance control in addition to a set of labyrinth seals or grooves **59** (FIG. **7**) cut into the outer surfaces of the bearing shells in the unloaded zone. Additional lubricant is admitted to bearing pressure pads in the bearing load zone at the inner diameter bearing surface from the high pressure plenum between the bearing and the housing.

All of this bearing drain flow is gathered at the ends of the bearings furthest from the cam member **14**. The drain drawing flow from the bearing at the drive end of the pump is directed through the main drive spline **22** to provide lubrication in that critical area. The drain flow for both bearings **15** and **16** is thus collected in one location at the boost end of the pump where it is returned, via cored passages **36** to the vane stage inlet. Some additional lubricant is permitted to flow from the boost end gathering point through the splines of the drive shaft **24** and ultimately drains to the area between the axial inducer and the impeller, this location chosen to assure that the hot drain flow cannot corrupt the capabilities of the boost stage **17**.

With reference to FIGS. **6** and **7**, the journal bearings **15** and **16** are a "hybrid" configuration incorporating the principles of both hydrodynamic and hydrostatic lubrication. A pressure-fed lubrication groove **59** is provided to feed the high pressure lubricant to the bearing. A pressure pad is formed from an axially Oriented groove **100** and a "U" shaped groove **101**. The axial groove **100** is supplied with high pressure lubricant through a feed hole **102** from the external groove **59** and its purpose is to provide spillover lubrication into the pad as well as provide a high reference pressure for increased load carrying capability. The "U" shaped groove **101** is supplied with high pressure lubricant through feed holes **103** and its purpose is to provide the high pressure reference around the remainder of the pad for increased load carrying capability. The grooves are not connected in order to assure that the spillover lubrication must occur and that the lubricant cannot be shunted through the U-groove away from the load zone. This hybrid configuration permits a lubricant film thickness which is substantially greater than that which could be achieved, under the same unit bearing loads, with a hydrodynamic configuration but which does not incorporate the high parasitic leakages which would occur with a pure hydrostatic bearing. The bearing drain pressure is referenced to boost stage discharge and thus assures sufficient ambient pressure to prevent bearing cavitation.

The bearings **15** and **16** are carefully suspended to assure that they will retain intimate proximity with the cam face and will remain stable throughout the operating range for the pump's entire operating life. One of the bearing blocks such as **15** is "grounded" within the housing and becomes the reference for the entire pump assembly. The cam **14** and the remaining bearing **16** are assembled relative to the bearing block **15**. Springs load against the end of the bearing block **16** which is furthest away from the cam **14** to assure intimate proximity of the three parts during initial start up. As fluid pressure is developed it applies force against the bearing flange **62** to increase the load of the bearing against the cam. A relief groove **101** allows low inlet pressure to bear against a substantial portion of the face of the bearing **16** which is adjacent to the cam **14**, to help assure that pressure loads will tend to clamp the bearings **15** and **16** to the cam **14**.

One end of the main drive shaft **21** incorporates a male spline **22** which engages with the engine gear box and is lubricated with engine gear box oil. The opposite end of the shaft also incorporates a male spline **23** which engages a matching female spline in the main pump rotor **12**. This spline is lubricated with fuel which is flushed through it as part of the internal flow schematic illustrated in FIG. **1**. The boost stage drive shaft **24** engages the same female spline in the main pump rotor **12** while the opposite end of the boost shaft is splined to engage the boost stage inducer section **18**.

All of the components of the present pumps are enclosed in cast aluminum housing sections **19** and **20**. The main vane stage is grounded through the bearings **15** and **16** against a housing structure which is designed to be very rigid yet light in weight, thereby assuring that none of the components of the vane pump cluster will become misaligned during high pressure operation. The housing material is selected for this application to be well suited for the fuel temperature range expected with a well established fatigue stress background.

It should be understood that the foregoing description is only illustrative of the invention. Various alternatives and modifications can be devised by those skilled in the art without departing from the invention. Accordingly, the present invention is intended to embrace all such alternatives, modifications and variances which fall within the scope of the appended claims.

What is claimed is:

1. A durable, single action, variable displacement vane pump capable of undervane pumping comprising:

(a) a cylindrical rotor member having journal ends and a central vane section comprising a plurality of radial vane slots uniformly spaced around the central circumference thereof, said vane slots being elongate in the axial direction and each having a central vane-supporting portion surrounded at each end by slot extension portions;

(b) a plurality of vane elements, each slidably-engaged within the central vane-supporting portion of a said vane slot for radial movement therewithin;

(c) a unitary cam member having opposed faces and a circular bore therethrough forming a cam chamber having a continuous interior cam surface, the central vane section of said rotor member being supported

axially and non-concentrically within said cam chamber so that the outer tip surfaces of all of the vane elements make continuous contact with said continuous interior cam surface during rotation of said rotor member, and said vane slot extensions project axially-outwardly beyond the faces of said cam member;

(d) an opposed pair of manifold bearings rotatably supporting the journal ends of said rotor member and overlying said vane slot extensions, each said bearing having a bearing face surface which contacts a face surface of said cam member and encloses the central vane-supporting portion of said rotor member within said cam chamber, each manifold bearing comprising an inlet arc segment containing means for admitting fluid to expanding vane bucket areas of the rotating vaned rotor, and means for admitting fluid into said vane slot extensions and undervane areas, and a discharge arc segment containing means for discharging pressurized fluid from contracting vane bucket areas of the rotating vaned rotor and from undervane areas as the vanes are depressed into the vane slots during rotation through the discharge arc,

said cam member being adjustable relative to said vaned rotor to vary the extent of eccentricity therebetween for varying the displacement capacity of said vane pump.

2. A vane pump according to claim **1** in which each face of the cam member contains inlet means adjacent an arcuate segment of the cam bore, corresponding to the inlet arc of the bearing faces, to admit inlet fluid to the expanding vane bucket areas.

3. A vane pump according to claim **1** in which at least one of said manifold bearings further includes:

an axial pressure groove having an inlet for pressure-fed lubricant providing pressure bias for the rotor in the incoming rotor direction; and a cooperatively positioned substantially U-shaped lubricating groove independent of said axial pressure groove and having an axial base portion and transversely positioned leg portions each having an inlet for pressure-fed lubricant; the said base portion being located in the outgoing rotor direction relative to said axial pressure groove.

4. A vane pump according to claim **1** in which said rotor member comprises a cylindrical barstock of relatively-uniform diameter having journal ends of said diameter.

5. A vane pumping according to claim **1** in which said rotor member further includes depressions in the rotor surface between said radial vane slots which provide additional fluid volume to reduce the effects of rapid pressure build-up during operation of the pump.

6. A vane pump according to claim **1** in which said central vane section comprises a plurality of radially-extending teeth, adjacent pairs of said teeth being formed as wall extensions of said vane slots to further support said vane elements during their radial movement within the vane slots.

7. A vane pump according to claim **1** in which each said vane slot has an arcuate floor which tapers uniformly from the central maximum depth portion upwardly and outwardly to said extension portions.

8. A vane pump according to claim **1** in which each vane slot has a contoured floor and each vane element has an undersurface which is contoured to correspond with the contour of the floor of the vane slot.

9. A vane pump according to claim **1** in which each said vane slot has an arcuate floor and the undervane face of each said vane is arcuate.

10. A vane pump according to claim **1** in which each bearing face also contains seal arc segments at transition areas between the inlet arc and the discharge arc segments, said seal arc segments having a sealing face for isolating the vane bucket areas from inlet and discharge pressures, and an inner diameter passage for opening the vane slot extensions and undervane areas to a source of fluid at a regulated pressure intermediate said inlet and discharge pressures.

11. A vane pump according to claim **10** in which each bearing face comprises an inlet arc of about 180°, a seal arc of about 36°, a discharge arc of about 108° and a second seal arc of about 36°.

12. A vane pump according to claim **1** in which each said vane slot contains a stop member which limits the extent of depression of the vanes into the vane-supporting portions of the slots and provides an undervane area for pressure-balancing and undervane pumping purposes.

13. A vane pump according to claim **12** in which said stop member comprises a raised floor portion, adjacent a deeper floor portion providing said undervane area.

14. A vane pump according to claim **1** in which each said manifold bearing has a bearing face surface comprising a major inlet arc segment, a minor discharge arc segment and smaller seal arc segments as transitional segments spacing said inlet and discharge arc segments, and passage means through each said bearing in said seal arc segments for communicating the vane slot extensions of the rotor member with a source of fluid pressurized to a predetermined intermediate pressure.

15. A vane pump according to claim **14** in which the said passage means through said manifold bearings in said seal arc segments are configured to produce substantially symmetrical forces on said unitary cam member throughout the range of adjustment of said cam relative to said vaned rotor.

16. A vane pump according to claim **14** further including a piston adjustment system for adjusting said cam relative to said rotor, wherein said piston adjustment system is actuated by fluid pressure supplied by said source of fluid pressurized to a predetermined intermediate pressure.

17. A vane pump according to claim **14** in which each said manifold bearing has a major inlet arc segment comprising a face surface having a plurality of relatively wide radial inlet recesses spaced by a plurality of relatively narrow stand-off face members, said inlet recesses opening axially into a common inlet chamber having an undervane inlet port at the inner diameter of said bearing.

18. A vane according to claim **14** in which each said manifold bearing has a minor discharge are segment comprising a face surface having axial openings to a discharge chamber having an undervane inlet port at the inner diameter of said bearing and having a discharge port at the outer diameter of said bearing for discharging pressurized fluid from the vane pump.

19. A vane pump according to claim **14** in which each said manifold bearing has a discharge arc segment in the face surface thereof bearing axially against said cam, having relief openings to the exterior for reducing the total pressure-induced force acting on said face, and said bearing further comprises a flange shoulder surface, axially opposite said face surface, that is subjected to pressure-induced force greater than the pressure-induced force acting on said face surface, for enhancing the seal between said cam and said manifold bearings.

* * * * *

United States Patent [19]
Anderson

[11] **3,759,588**
[45] **Sept. 18, 1973**

[54] **HIGH SPEED HYBRID BEARING COMPRISING A FLUID BEARING & A ROLLING BEARING CONNECTED IN SERIES**

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[22] Filed: **Nov. 12, 1971**

[21] Appl. No.: **198,379**

[52] **U.S. Cl.** **308/35, 308/9**
[51] **Int. Cl.** **F16c 21/00, F16c 39/04**
[58] **Field of Search** .... 308/35, 9, 160, DIG. 1

[56] **References Cited**

UNITED STATES PATENTS

3,143,381	8/1964	Vurpillot	308/9
1,445,188	2/1923	Wadsworth	308/160
1,175,415	3/1916	Egbert	308/35
2,623,353	12/1952	Gerard	308/35
3,012,827	12/1961	Goetz	308/35
3,026,154	3/1962	Marchand	308/35

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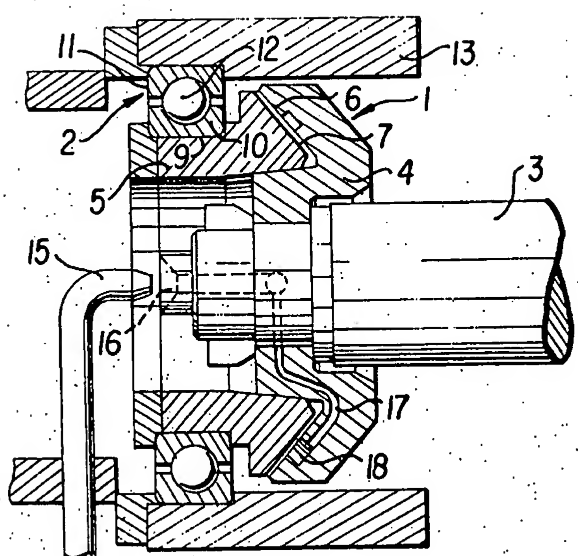
[57] **ABSTRACT**

A rotating shaft is supported by a fluid bearing and a rolling element bearing coupled in series. Each bearing turns at a fraction of the rotational speed of the shaft. The fluid bearing is preferably conical, thereby providing thrust and radial load support in a single bearing structure.

7 Claims, 3 Drawing Figures



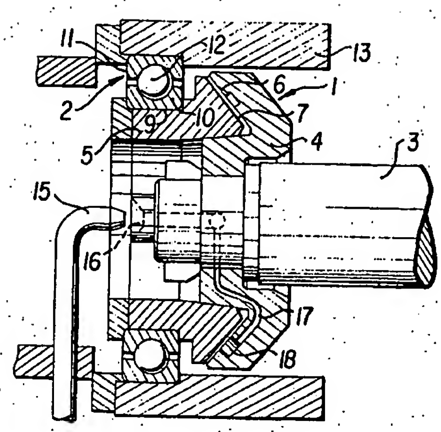
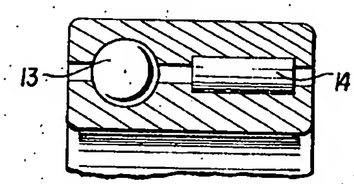


FIG. 1

FIG. 2



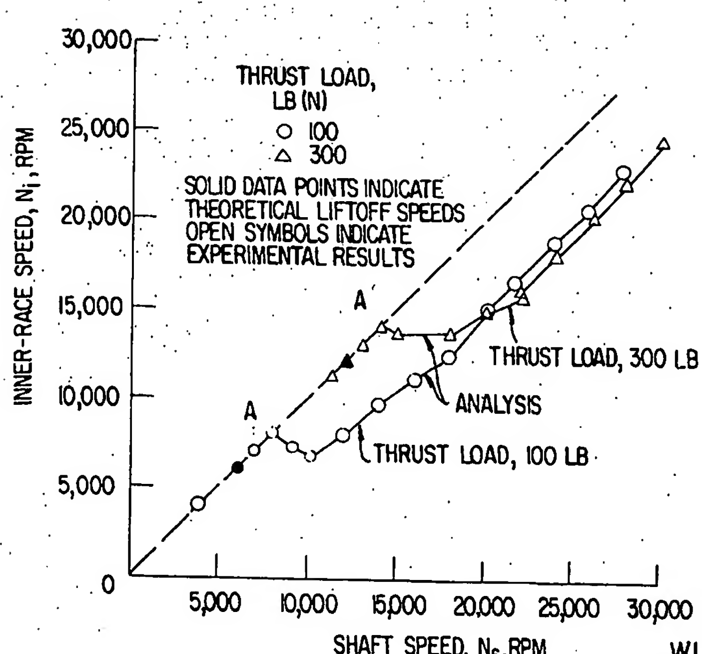


FIG. 3

INVENTOR
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BY Norman T. Musial
Gene E. Shook ATTORNEYS

HIGH SPEED HYBRID BEARING COMPRISING A FLUID BEARING & A ROLLING BEARING CONNECTED IN SERIES

ORIGIN OF THE INVENTION

The invention described herein was made by employees of the United States Government and may be manufactured and used by or for the Government for governmental purposes without payment of any royalties thereon or therefor.

BACKGROUND OF THE INVENTION

1. Field of Invention

The invention is related to bearings, more specifically, to a hybrid bearing in which a rotating shaft is supported by a fluid film bearing and a rolling element bearing coupled in series in a manner that each turns at a fraction of the shaft speed.

2. Description of the Prior Art

Recent developments in gas turbine engines—such as higher thrust-to-weight ratios, advanced compressor design, high temperature materials, and increased power output—have necessitated larger shaft diameters and higher main shaft bearing speeds for future engine designs. For example, bearings in current production aircraft turbine engines operate in the range from 1.5 to 2 million DN (bearing bore in mm times and shaft speed in rpm). However, engine designers anticipate that turbine bearing DN values will have to increase to the range of 2.5 to 3 million in the near future and further developments may require bearing DN values as high as 4 million after 1980.

These higher values create significant wear problems since when ball bearings are operated at DN values above 1.5 million, centrifugal forces increase Hertz stresses at the outer-race ball contacts and seriously shorten bearing fatigue life. At typical aircraft thrust loads carried by bearings, such as 2,000 and 4,000 pounds, an increase from a DN of 1.8 million to 4.2 million results in a reduction in life of 98 and 96 percent, respectively.

Many attempts have been made to reduce the centrifugal forces and thus increase the life of high speed bearings. One method of accomplishing this result is to decrease the mass of the rolling elements by using hollow or drilled elements. The technique has resulted in improving fatigue life by a factor of 2.5 at a DN of 3 million and thrust load of 4,000 pounds.

A second technique for diminishing centrifugal forces utilizes hybrid bearings to eliminate the use of rolling bearings at high speeds. These bearings generally contain a rolling element bearing and a fluid bearing. The former is used for starting, stopping and low speed operation when centrifugal forces are minimal. The latter operates only at high speeds. Various devices are used to achieve this alternate operation. A clutch activates a fluid bearing in Banerian U.S. Pat. Nos. 2,986,430 and 3,058,786. Centrifugal force disengages ball bearings in Hiatt et al. U.S. Pat. No. 3,360,310 and Marchand U.S. Pat. No. 3,026,154. Goetz U.S. Pat. No. 3,012,827 illustrates the use of ball bearings to support a rotating shaft before a gas bearing begins its support function.

The prior art hybrid bearing significantly increases bearing life since the rolling element bearing ceases to operate at the higher shaft speeds when the entire rotational load is supported by the fluid bearing. A disadvantage of hybrid bearings of this design is their large power losses compared with conventional ball bearing which results from the relative inefficiency of the fluid bearing. High power loss makes them unsuitable for some applications.

McNaughton et al. U.S. Pat. No. **2,872,254** discloses a device similar in operation to these hybrid bearings. However, in contrast to the other prior art devices which are concerned with high speed bearing applications, McNaughton's device is useful to achieve operation over a wide temperature range. This hybrid includes a ball bearing element packed with a lubricant having a high viscosity at room temperature and a sliding element. Heat generated by the sliding element, which initially supports the shaft, decreases the viscosity of the ball bearing lubricant and causes the sliding element to lock, thus transferring the rotation to the ball bearings. If the sliding element does not rigidly lock, some speed sharing between the bearings occurs; however, it is minimal, uncontrolled and unpredictable.

Composite bearings in which the load is shared by the various bearing elements connected in parallel are also known. In this arrangement each of the bearing elements turns at the shaft speed. Some increase of bearing life is attained through load sharing, but at high DN values. This is less than that attained through speed sharing. Examples of parallel composites are disclosed in U.S. Pat. Nos. 2,875,001; 3,065,036; 3,301,611 and 3,305,280.

SUMMARY OF INVENTION

An object of this invention is therefore to provide a bearing containing a rolling element in which fatigue life is substantially increased while maintaining a level of power loss no greater than that of an equivalent conventional rolling element bearing.

A second object of this invention is to reduce the stresses on the outer race contacts of a rolling element bearing caused by centrifugal forces generated at high shaft speeds.

Another object of this invention is to provide a composite bearing in which each bearing rotates at a fraction of the shaft speed.

A further object of this invention is to provide a composite bearing composed of a fluid film bearing and a rolling element bearing connected in a manner that the inner race of the latter bearing rotates at less than the shaft speed.

A still further object of this invention is to provide a composite bearing which will support both radial and thrust loads.

These and other objects are accomplished by providing a composite bearing which comprises a fluid film bearing and a rolling element bearing connected in series in a manner which reduces the orbital speed of the rolling element by decreasing the speed of the inner race of the rolling element bearing to a fraction of the shaft rate. This is accomplished by connecting the first element of the fluid film bearing to a shaft and the second element to the inner race of the rolling element bearing in a manner that the latter two elements rotate at the same speed. The outer race of the rolling element bearing is mounted on a stationary housing. In the context of this application the inner race is defined as the one attached to the fluid bearing element; the outer race as the one attached to a stationary support. In operation the first element turns at the full shaft speed

and the second element and the inner race rotate at a slower rate.

Since the DN value is a function of bearing bore and shaft speed, it is clear that its value is not decreased by the hybrid bearing of this invention. However, it is equally clear that the fatigue life of the bearing will be increased because the speed of the rolling element in this bearing is less than the speed of the rolling element in a conventional roller bearing at a given DN. The benefits of the invention are illustrated by comparing fatigue life improvement obtained with a hybrid bearing using a ball bearing, assuming a 30 percent reduction in roller element speed, with that of hollow ball bearings. At a 4,000 pound thrust and DN values of 3 and 4 million bearing life should theoretically improve by a factor of 3.2 and 5.7, respectively, over that of conventional bearings using solid balls. In contrast under the same conditions fatigue life using hollow balls was improved only by a factor of 2.5 and 4.2. The theoretical values for the bearings of this invention were obtained by comparing fatigue life values for solid ball bearing with those obtained at a DN of 30 percent less.

DESCRIPTION OF THE DRAWINGS

FIG. 1 illustrates a cross section of the composite bearing of this invention;

FIG. 2 is a cross section of the antifriction element of the subject invention wherein the rolling element is a combination roller and ball bearing;

FIG. 3 graphically illustrates the speed sharing characteristics of the present invention.

DESCRIPTION OF THE PREFERRED EMBODIMENTS

FIG. 1 illustrates the subject invention which comprises a fluid film bearing 1 and a rolling element bearing 2 connected in series journaled to a rotating shaft 3. The fluid bearing is of the conventional type and, comprises first and second elements 4 and 5 containing first and second opposed surfaces, 6 and 7, respectively. As in conventional fluid bearings sufficient clearance is provided between the surfaces to permit the second surface to float and slide freely on a fluid film maintained between said first and second surfaces. The surfaces may be parallel or perpendicular to the axis of the journaled shaft. In the former configuration the bearing will support only radial loads; only thrust loads will be supported in the latter case. Alternatively, the bearing may have four surfaces, two aligned parallel to the axis and two perpendicular, thus supporting axial and thrust loads. This dual support function can also be accomplished in the preferred arrangement shown in FIG. 1 wherein the two bearing surfaces are conical in shape having their apex at the shaft axis.

The first bearing element has a central hole for journaling. The second contains a means for mounting the inner race of the rolling element bearing. In FIG..1, this means is shown as a groove 9.

The rolling element bearing is composed of a first and second race, 10 and 11, respectively, having a plurality of rolling elements 12 between them. In FIG. 1, the rolling element is shown as a ball bearing, however, any conventional roller or ball and roller combination bearing can also be used. In FIG. 2, a combination roller element bearing containing ball and roller elements 13 and 14, respectively, is shown. The inner race 10 of the roller bearing is mounted in the groove 9 contained on the second element of the fluid bearing. The outer race 11 is mounted on a stationary housing 13.

The fluid bearing may be lubricated by any conventional means. In the device illustrated in FIG. 1 a jet 15 guides oil into an opening 16 in the shaft 3. The oil flows through a canal 17 to a lubricant supply orifice 18 and then flows between the bearing elements. Centrifugal forces generated by the rotation of the shaft pumps the oil to the bearings. Alternatively, the fluid bearing can be lubricated independently of the rotation of the shaft.

A rotating shaft which is journaled in the hybrid bearing of the present invention is initially supported only by the rolling bearing. That is, the inner race of the rolling bearing rotates at the full shaft speed. As shaft speed increases, sufficient lubricant pressure is built up to start the fluid bearing rotating. At this point the inner race speed drops abruptly, indicating that the fluid bearing is now rotating at a portion of the shaft speed. The inner race and the second fluid bearing element are then rotating at the same rate which is less than the shaft speed.

FIG. 3 graphically illustrates the operation of the subject invention under 100 and 300 pound thrust loads, where point A designates the lift off of the fluid bearing. In the figure the dashed line illustrates that in a conventional ball bearing the inner race speed is the same as the shaft speed. It can be seen that at low shaft speeds the hybrid bearing of this invention operates similarly to the conventional bearing. (Of course this is not true if the fluid bearing is pressurized.) However, when sufficient shaft speed is attained the fluid bearing lifts off and the inner race speed drops. Note that the liftoff speed is higher for the 300 pound load since higher fluid pressures are necessary to float the fluid bearing. It is clear from these curves that the inner race speed of the ball bearing of the hybrid bearing is less than those in conventional ball bearings for the same shaft speed. This decrease results in significant increases in bearing fatigue life.

I claim:

1. A hybrid bearing comprising
   a fluid film bearing comprising first and second elements
   first and second substantially parallel and opposed surfaces on said first and second elements respectively, and
   a rolling element bearing including an inner and an outer race, one of said races being mounted on said second element whereby said rolling element bearing is connected in series with said fluid bearing so that said second element rotates at a fraction of the speed of said first element when the driving torque through said fluid film bearing equals the restraining torque of said rolling element bearing.

2. A hybrid bearing according to claim 1, wherein said fluid film bearing surfaces support both radial and axial loads.

3. A hybrid bearing according to claim 2, wherein said fluid film bearing surfaces are conical in shape.

4. A hybrid bearing according to claim 1, wherein said rolling element bearing is a ball bearing.

5. A hybrid bearing according to claim 1, wherein said rolling element bearing is a combination ball and roller bearing.

6. A hybrid bearing comprising

a rotatable first element having a first smooth surface thereon,

a second element having a second smooth surface aligned in a substantially parallel and opposed relationship with said first smooth surface,

means for forming a fluid film between said first and second surfaces,

an inner race secured to said second element remote from said second surface,

a fixedly mounted outer race surrounding said inner race in spaced relationship, and

a plurality of rolling elements interposed between said inner and outer races so that when the driving torque through the fluid film equals the restraining torque of said rolling elements the second element rotates at a constant speed which is less than the rotational speed of said first element.

7. A hybrid bearing according to claim 6, wherein said substantially parallel opposed surfaces are conical in shape having their apices at the axis of rotation of said bearing.

* * * * *

United States Patent [19]

Smith

[11] **Patent Number: 4,927,274**

[45] **Date of Patent: May 22, 1990**

[54] **SLIP RING AIR BEARING**

[76] Inventor: **Robert S. Smith**, 1263 Emory St., San Jose, Calif. 95126

[21] Appl. No.: **370,607**

[22] Filed: **Jun. 23, 1989**

[51] Int. Cl.[5] **F16C 21/00**

[52] U.S. Cl. **384/101**; 384/102; 384/126; 384/624

[58] Field of Search 384/100–102, 384/114, 126, 624

[56] **References Cited**

U.S. PATENT DOCUMENTS

3,305,280	2/1967	Peterson	384/126
3,491,529	1/1970	Kerr et al.	384/100 X
3,642,331	2/1972	Silver	384/102
3,751,044	8/1973	Sawada	384/101 X
3,854,781	12/1974	Bildtsén	384/102

FOREIGN PATENT DOCUMENTS

2359389	6/1975	Fed. Rep. of Germany	384/624
1287595	2/1962	France	384/114

Primary Examiner—Thomas R. Hannon
Attorney, Agent, or Firm—Robert Samuel Smith

[57] **ABSTRACT**

A compound journal airbearing having a sleeve concentric with a rotor and having an inner surface forming an airbearing space with the surface of the rotor so as to maintain the rotor surface and inner sleeve surface out of contact. The sleeve is supported at each end by the inner race of a bearing whose outer periphery is supported by contact with the interior surface of the stator housing. Pressurized air is supplied to the airbearing space by a passage leading from an air supply through the wall of the stator and sleeve to the airbearing space. The airbearing surface can be shaped to provide both radial and axial support. For this purpose, the airbearing surfaces may be either concave or convex toward the axis of the rotor. Catastrophic failure that would occur with construction of the prior art due to inadvertent contact of rotor and stator surfaces (collapse of the airbearing film) is avoided because the sleeve is able to turn when inadvertent touchdown occurs.

7 Claims, 2 Drawing Sheets

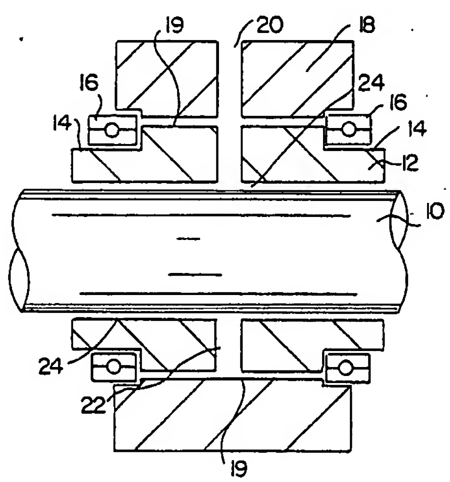


FIG. 1

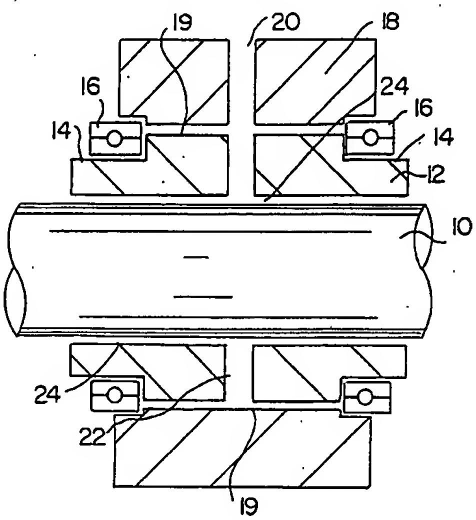


FIG. 2

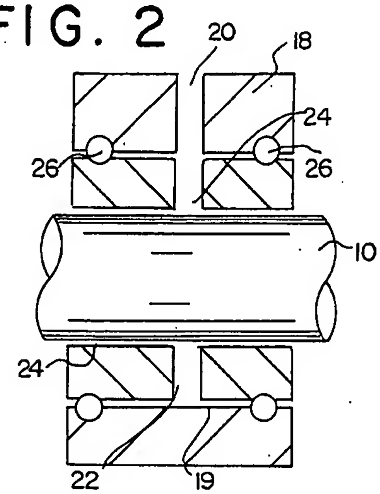


FIG. 3

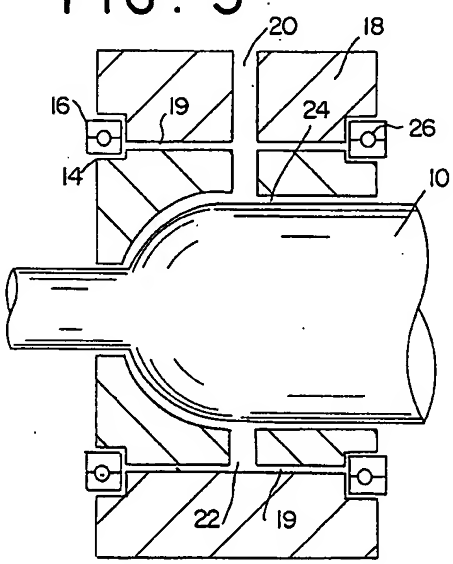
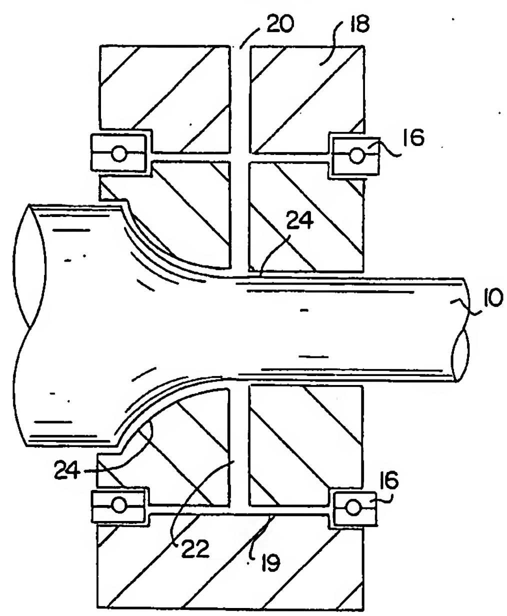


FIG. 4



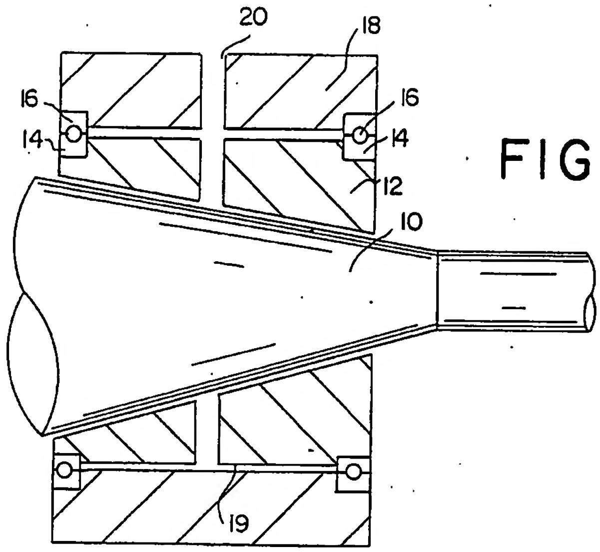


FIG. 5

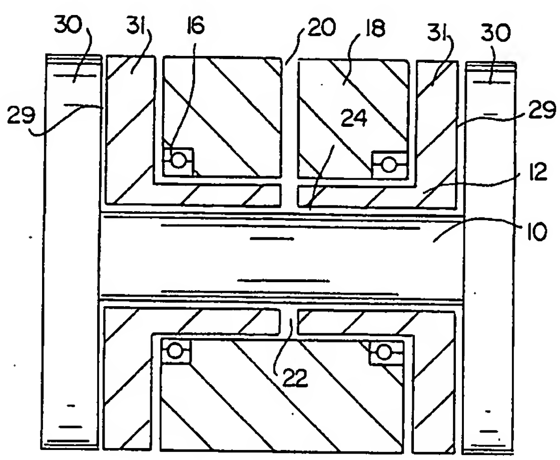


FIG. 6

SLIP RING AIR BEARING

BACKGROUND

1. Field of the Invention:

This invention is related to journal air bearings and in particular to an air bearing having a slip ring construction that minimizes the problem of galling between the air bearing surfaces.

2. Prior Art:

Journal air bearings are bearings comprising a cylindrical rotor enclosed by a stator (or housing) in which the surface of the rotor is supported out of contact with the interior surface of the stator by a pressurized air film. The air film is the lubricant for the bearing and therefore a major advantage of this construction is the low frictional loss that is associated with the bearing. A second advantage is the minimal transmission of vibration from the stator to the rotor. Journal air bearings are therefore used in applications where elimination of vibration is very important such as to support the spindles of latches or grinders used to manufacture substrates for memory disks. Journal air bearings may be designed to support the rotor in both the radial and axial directions. Axial support is provided by a flange on each end of the rotor that is parallel and in close proximity to a stationary flange on each end of the stator. The interface between the rotor and stator flanges is an airbearing space, wherein pressurized air is supplied through a passage leading from a source of pressurized air, through the wall of the stator and opening into the airbearing region. Other constructions are also used to provide greater support depending on the application. These constructions include spherical interfaces which in one case is convex toward the axis of the bearing and in another case is concave toward the axis of the bearing. In another construction, the rotor is tapered.

The performance of any air bearing depends on the distribution of pressure and pressure gradients in the air bearing space. These distribution patterns are established by shallow grooves in the surface of the stator bounding the air bearing space and lead from the proximity to the entry of the air passage into the airbearing space to various areas of the interface.

For a detailed discussion of the prior art, reference is made to "Hydrostatic and Hybrid Bearing Design" by W. B. Rowe, published by Butterworth & Co., Univ. Press, and available in the Library of Congress, TJ1073.5.R69 1983.

A major problem with journal air bearings is their susceptibility to "crashes". A "crash" is the term applied to a situation where, for one of a number of reasons, oftentimes a speck of dirt, momentary contact between the airbearing surfaces occur leading to collapse of the air film, severe galling of the two surfaces moving in frictional contact with one another and sudden "freezing" of the rotor. When a crash occurs, the rotor must be removed from the stator, the surfaces lapped and polished, and the bearing reassembled. Usually the lapping must be performed by the manufacturer of the air bearing so that considerable time and effort is expended.

The traditional approach to avoiding crashes has been to pump very clean (filtered) air through the air bearing. Filters are normally designed to prevent only a fraction of particles larger than a given value from passing. Therefore the problem persists because it is virtually impossible to prevent particles from eventually passing through the filtering system into the airbearing space.

In order to minimize the damage due to the crashes, airbearing surfaces are made that are very hard. Surfaces that are used are "hard anodized" aluminum, and nitrided (case) hardened steel. The problem here is that preparation of these surfaces is costly and critical. Furthermore, when crashes occur involving these surfaces, the relapping that is necessary often "breaks through" the hardened surface layer and the parts must be retreated or discarded.

Another approach to minimize damage due to a crash has been to install a relatively expensive braking system that stops the rotation within a period of one revolution of the beginning of a crash. The latter remedy is obviously not effective by definition.

THE INVENTION

OBJECTS:

It is an object of this invention to provide a journal air bearing whose construction prevents "crashing".

It is a further object that intermittent touching of the air bearing surfaces of this construction will not lead to the crashes and galling of the airbearing surfaces.

It is another object of this invention to incorporate a versatile character to the construction of the bearing that avoids the expense of reconstruction and repair of crashed air bearings such as are required by the constructions of the prior art.

SUMMARY:

This invention is directed toward a journal air bearing having a sleeve that encloses a rotor wherein the adjacent surfaces of rotor and sleeve are maintained out of contact by an airbearing film and the sleeve is supported in a housing that permits the sleeve to turn when there is the slightest rotational drag on the sleeve such as might occur with intermittent contact of the sleeve and rotor.

In one embodiment, the sleeve is supported at each end by the inner race of a bearing which is supported in turn at its periphery by contact with the interior surface of a (stator) housing. A passage through the housing supplies pressurized air to the air bearing area.

In another embodiment, the sleeve is supported within the cylindrical cavity of the housing by an "0" ring on each end of the sleeve. In this construction, lubricant may be supplied to the interface between the adjacent surfaces of the sleeve and interior surface of the housing.

For some applications, axial support of the bearing is presented by constructions at the end of the rotor that are unassociated with the bearing. In such situations, a sleeve having a cylindrical interior surface may be used. However, other shapes of the sleeve may be used to provide axial support. For example, the airbearing surface may be either concave or convex toward the axis of the rotor.

Since the airbearing surfaces are out of contact with one another and since the drag on touchdown results in negligible damage, materials having properties that occur in a wide range can be used to construct the rotor, sleeve and stator. For example, a preferred surface for the cylindrical interior of the housing is a hardened surface such as nitrided steel or hard anodized aluminum. Hard anodized aluminum has a naturally occurring porous surface that can be impregnated with agents

such as TEFLON plastic (fluorocarbon) so as to substantially reduce the coefficient of friction of the surface. The rotor may be made of a similar material.

The sleeve may be made of a material that has a low coefficient of friction and resists damage from adhesive wear that can occur from incidental contact by the rotor. If a relatively large air bearing spacing is to be used, which is the case when a large flow of air through the bearing can be permitted, dimensional stability and machining tolerance of the sleeve are not so critical and relatively soft materials such as polyvinyl chloride or phenolic can be used, such materials may absorb a foreign particle in the airbearing region and thereby avoid galling experienced with harder materials of the prior art. When only a small current of air is available requiring close machine tolerances, the sleeve may also be made of a hard material such as anodized aluminum with a surface having a low coefficient of friction as discussed above.

DRAWINGS:

FIG. 1 shows a cross sectional view of the slip ring airbearing of this invention featuring bearing support of the slip ring.

FIG. 2 shows a cross sectional view of the slip ring airbearing of this invention featuring "0" ring support of the slip ring.

FIG. 3 shows a cross sectional view in which the air bearing surfaces are convex toward the axis of the rotor.

FIG. 4 shows a cross sectional view in which the air bearing surfaces are concave toward the axis of the rotor.

FIG. 5 shows a cross sectional view in which the airbearing surfaces are tapered.

FIG. 6 shows a cross sectional view in which axial support to the rotor is provided by a flange on each end of the rotor.

DESCRIPTION OF A PREFERRED EMBODIMENT:

The following detailed description illustrates the invention by way of example, not by way of limitation of the principles of the invention. This description will clearly enable one skilled in the art to make and use the invention, and describes several embodiments, adaptations, variations, alternatives and uses of the invention, including what I presently believe is the best mode of carrying out the invention.

Turning now to a detailed discussion of the drawings, there is shown in FIG. 1 a cross sectional view of the slip ring airbearing of this invention. There is shown a rotor 10. A cylindrical sleeve 12 is positioned concentrically on the rotor 10. The sleeve 12 has a concentric shoulder 14 on each end. A bearing 16 is pressed onto each shoulder 14. The bearings are supported inside a stator 18 (housing enclosing a cylindrical cavity) so as to define an air passage space 19 between the outer surface of the sleeve and the inner surface of the stator. An air passage 20 conducts pressurized air from an air supply (not shown) through the wall of the stator 18 to air space 19. Air then passes from the airspace 19, through a plurality of radial channels 22 in the sleeve to the airbearing area 24 between the surface of the rotor and the inner surface of the sleeve. Grooves (not shown) in the interior surface of the sleeve establish the pattern of pressure and pressure gradients over the surface of the airbearing space. The required distribution of pressure depends on the application of the bearing and comprise, variously, recesses in the vicinity of the entrance of the air passage, grooves extending from the air passage entrance to the edge and around the edge of the airbearing space, etc. The procedure for determining the most effective layout of grooves is well known to those skilled in the art and is thoroughly discussed in the reference cited in the prior art.

Other means of supporting the sleeve may be selected other than a ball bearing. As illustrated in FIG. 2, an "0" ring 26 may be used on each end of the sleeve in place of the bearings and/or the interface between the sleeve and stator housing may be lubricated.

In FIGS. 3, 4, 5 and 6, constructions are shown which provide axial support to the rotor. In each fig. like numbers identify like parts. In FIG. 3, the airbearing space between the rotor and internal surface of the sleeve is convex toward the axis of the rotor. In FIG. 4, the airbearing space between the rotor and sleeve is concave toward the axis of the rotor. In FIG. 5, the air bearing space between the rotor and sleeve is tapered. In FIG. 6, a flange 4 on each end of the rotor forms an air bearing interface 29 with a surface of a stationary flange 31 on each end of the stator.

A preferred material for construction of the rotor and stator housing is aluminum having a hard anodized surface or steel having a nitrided surface. The material for fabricating the sleeve depends on the end use. For precise positioning of the rotor where the separation of the airbearing surfaces is less than 0.0005", the sleeve may be fabricated from aluminum and have a hard anodized surface. Another choice would be steel having a nitrided surface. If the airbearing space is designed to be large, then the material may be a plastic such as fiber glass filled epoxy.

The embodiments described in the foregoing paragraphs illustrate the means by which the objects of this invention are met. The slip ring construction is effective in avoiding the problem of "crashing" experienced with air bearings of the prior art. The construction is amenable to easy replacement of the sleeve when that is required.

It should be understood that various modifications within the scope of this invention can be made by one of ordinary skill in the art without departing from the spirit thereof. It therefore with my invention to be defined by the scope of the appended claims as broadly as the prior art will permit, and in view of the specification if need be.

I claim:

1. A journal air bearing to support a rotor having an axially symmetric surface, said bearing comprising:

- a sleeve having two ends and an inner and outer surface concentric with said rotor;
- said inner surface defining an air bearing space with said rotor surface;
- a stator having a stator wall which defines a chamber having a surface enclosing said sleeve;
- a ball bearing on each said sleeve end having an outer surface in supporting contact with said chamber surface and an inner surface in supporting contact with said said outer sleeve surface defining a closed space between said chamber surface and said outer sleeve surface;
- said airbearing space supplied by pressurized air from a supply of air through a passage leading from said pressurized air supply, through said stator wall and said sleeve, to said airbearing space.

2. A journal air bearing as in claim 1 wherein said inner sleeve and rotor surfaces are convex with respect to the axis of said rotor to provide both radial and axial support to said rotor.

3. A journal air bearing as in claim 1 wherein said inner sleeve and rotor surfaces are concave with respect to the axis of said rotor to provide both axial and radial support to said rotor.

4. A journal air bearing as in claim 1 wherein said inner sleeve and rotor surfaces are cylindrical.

5. A journal air bearing as in claim 1 wherein said inner sleeve and rotor surfaces are tapered.

6. A journal airbearing as in claim 1 which further comprises a flange attached to each end of said rotor and a flange fixed to each end of said sleeve, all flanges being perpendicular to said rotor axis thereby defining an airbearing space between each said flange attached to said rotor end and each said flange attached to said said sleeve end respectively wherein all said airbearing spaces communicate with said passage supplying pressurized air.

7. A journal airbearing as in claim 1 wherein said sleeve is a plastic.

* * * * *

United States Patent [19]

Simmons

[11] **Patent Number: 5,000,584**

[45] **Date of Patent: Mar. 19, 1991**

[54] **BUSHING FOR OIL FILM BEARING**

[75] Inventor: **Thomas E. Simmons**, Westborough, Mass.

[73] Assignee: **Morgan Construction Company**, Worcester, Mass.

[21] Appl. No.: **487,287**

[22] Filed: **Mar. 2, 1990**

[51] **Int. Cl.**[5] **F16C 32/06**; F16C 33/10

[52] **U.S. Cl.** **384/118**; 384/114; 384/291; 384/322

[58] **Field of Search** 384/100, 114–118, 384/120, 123, 283, 291, 293, 322, 397

[56] **References Cited**

U.S. PATENT DOCUMENTS

3,385,009 5/1968 Lueders .... 384/100 X
3,386,783 6/1968 Scheufler 384/291
3,453,031 7/1969 Rickley et al. 384/118
3,761,149 9/1973 Ferguson 384/114
4,159,152 6/1979 Bjork 384/291

FOREIGN PATENT DOCUMENTS

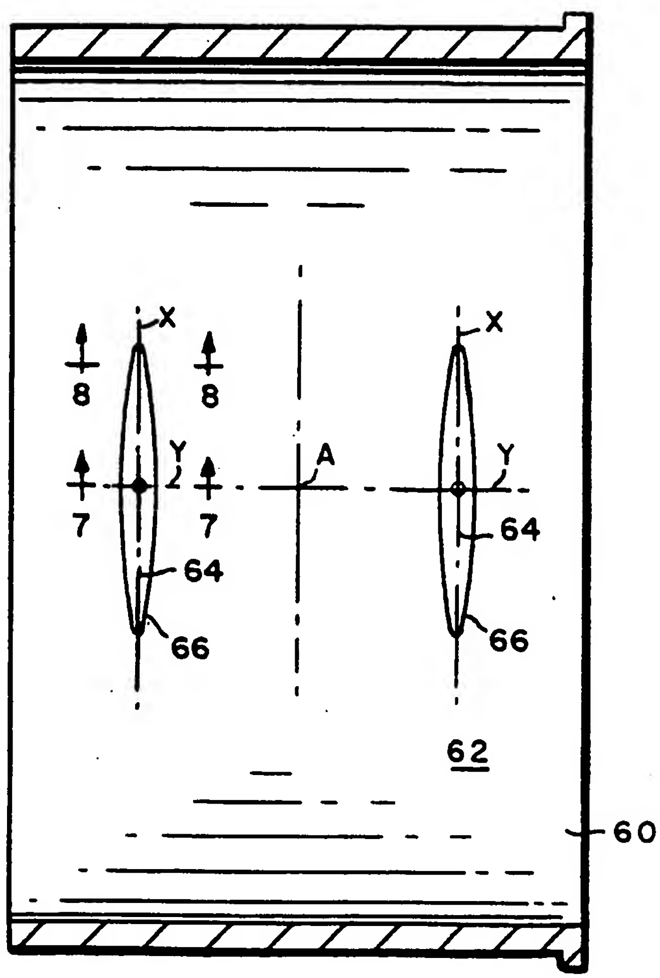
3825449 6/1989 Fed. Rep. of Germany 384/322

Primary Examiner—Thomas R. Hannon
Attorney, Agent, or Firm—Samuels, Gauthier & Stevens

[57] **ABSTRACT**

A bushing for an oil film bearing assembly has a plurality of hydrostatic oil recesses in its internal bearing surface at the load zone. Each recess is defined by the intersection of a double curved surface of revolution with the bearing surface, the recess thus being in the form of a portion of a prolate spheroid, with a substantially elliptical feathered peripheral edge.

10 Claims, 4 Drawing Sheets



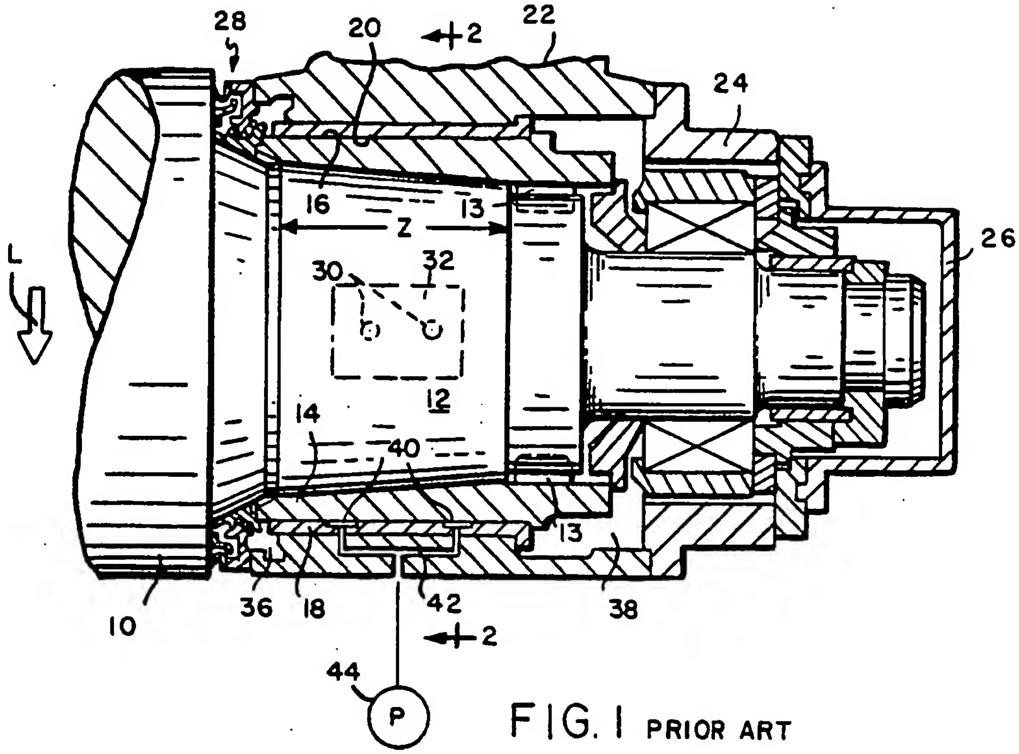


FIG. 1 PRIOR ART

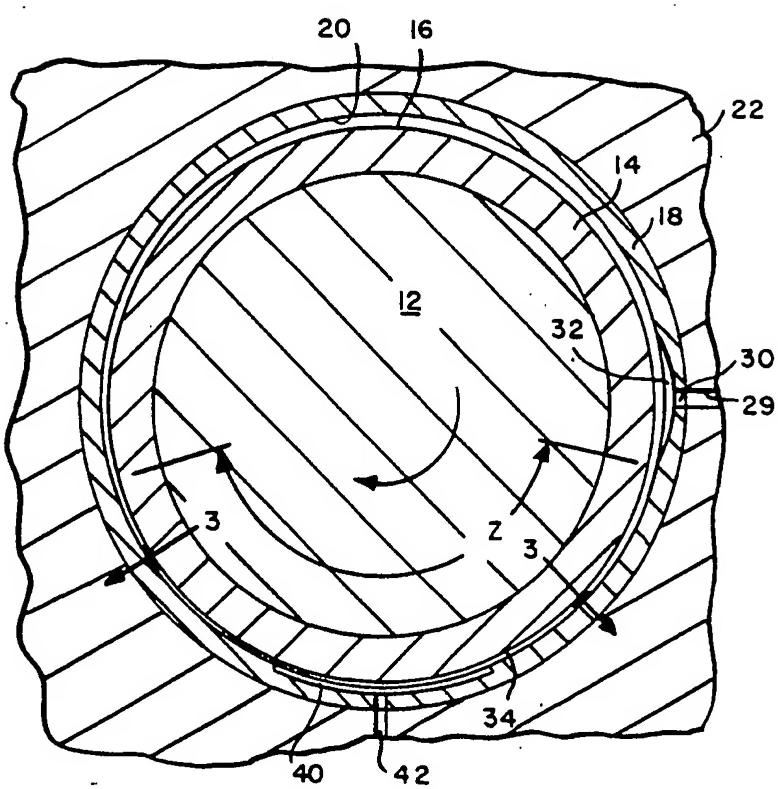


FIG. 2 PRIOR ART

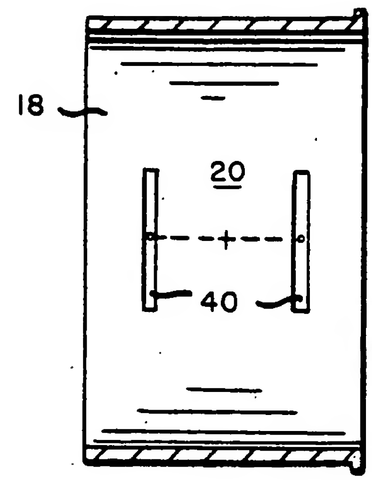


FIG.3 PRIOR ART

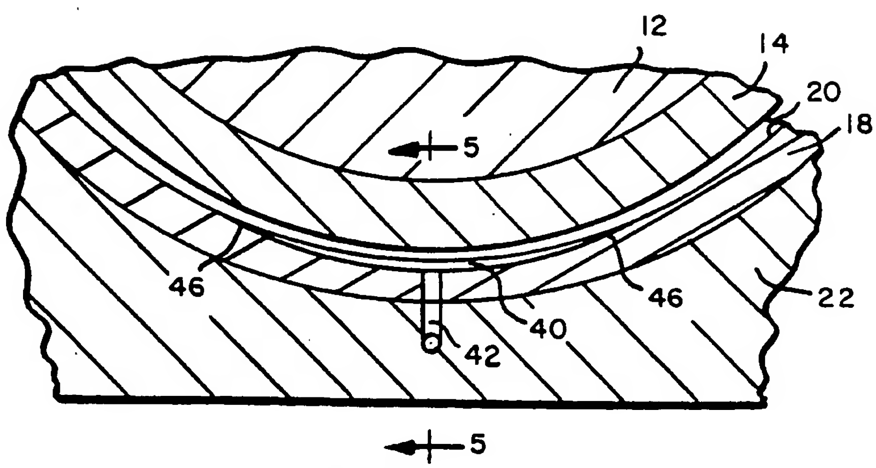


FIG.4 PRIOR ART

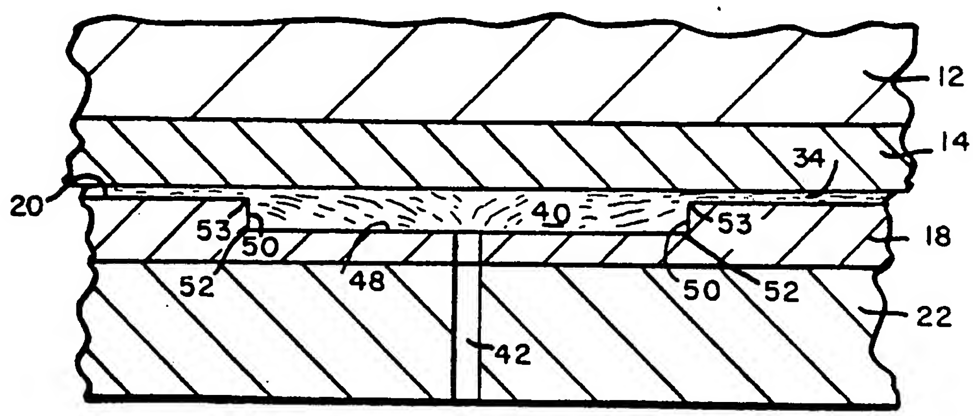


FIG.5 PRIOR ART

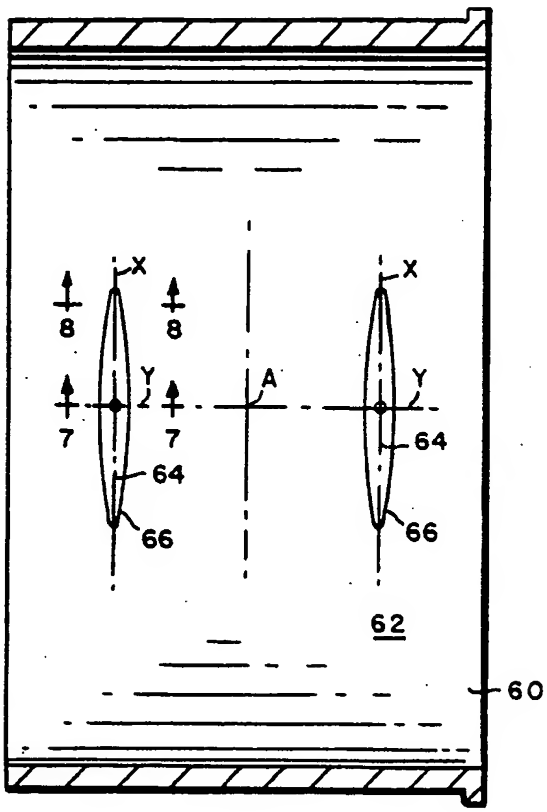


FIG. 6

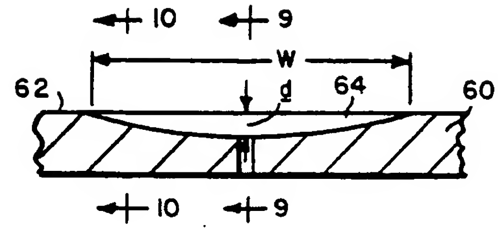


FIG. 7

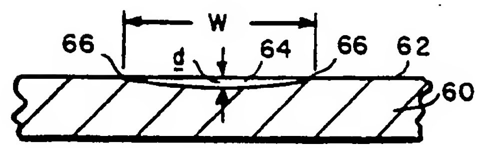


FIG. 8

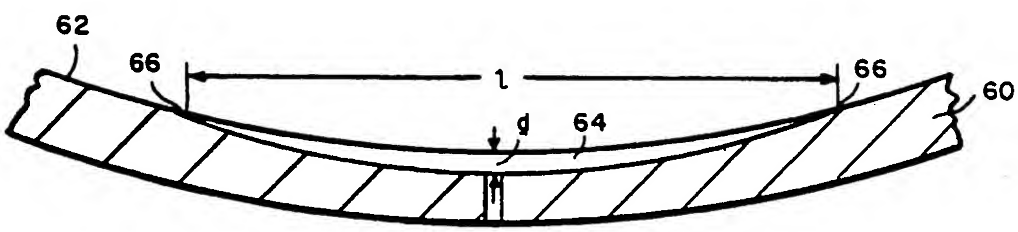


FIG. 9

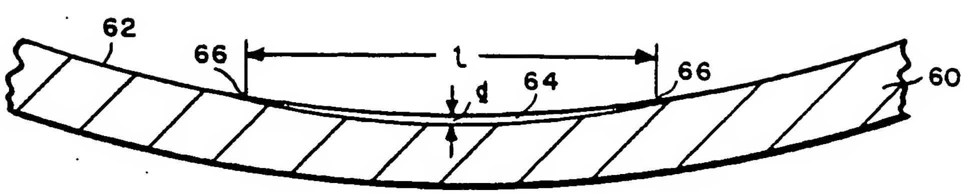


FIG. 10

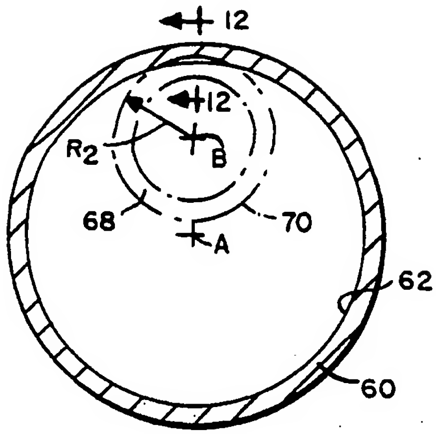


FIG. 11

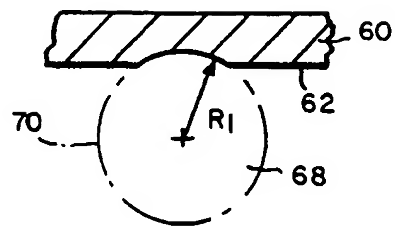


FIG. 12

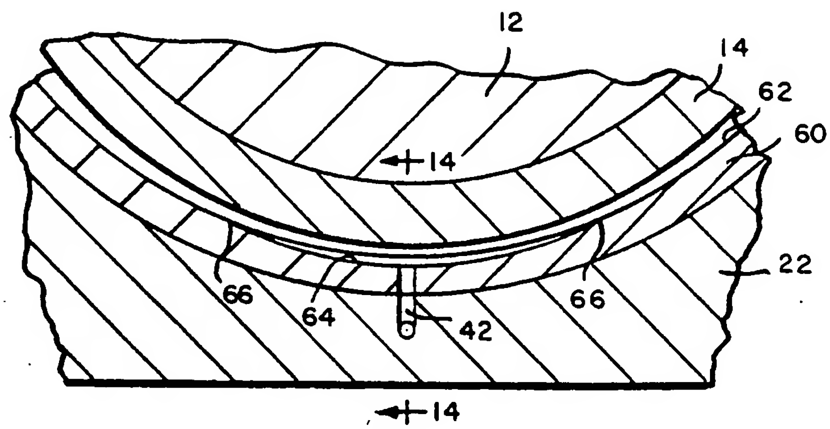


FIG. 13

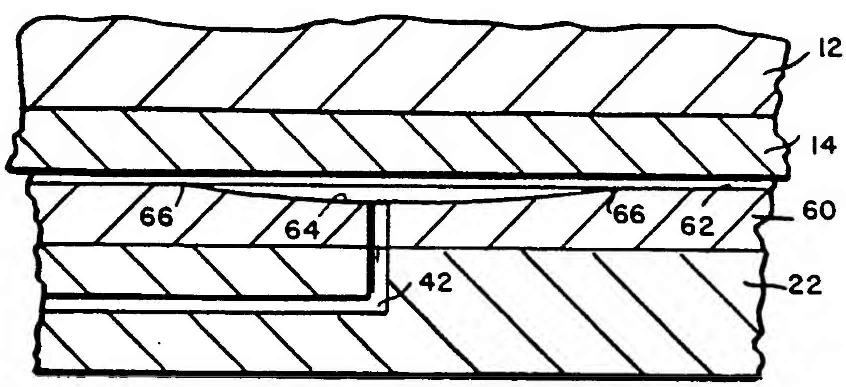


FIG. 14

BUSHING FOR OIL FILM BEARING

BACKGROUND OF THE INVENTION

1. Field Of The Invention

This invention relates to an improvement in oil film bearings of the type employed to rotatably support the journal surfaces of roll necks in a rolling mill.

2. Description of the Prior Art

In the typical rolling mill application of an oil film bearing, as depicted somewhat diagrammatically in FIGS. 1 and 2, the roll 10 has a neck section 12. The neck section 12 may be tapered, as shown, or it may be cylindrical. A sleeve 14 is received on the neck section 12 where it is rotatably fixed by means of one or more keys 13. The exterior of the sleeve defines the journal surface 16 of the roll neck. A bushing 18 has an internal bearing surface 20 surrounding and rotatably supporting the journal surface 16. The bushing is contained by and fixed within a chock 22. The chock is adapted to be supported in a roll housing (not shown), and is closed at the outboard end by an end plate 24 and cover 26. A seal assembly 28 is provided between the roll and the inboard end of the chock 22. The seal assembly functions to retain lubricating oil within the bearing while at the same time preventing contamination of the lubricating oil and inner bearing components by cooling water, mill scale, etc.

During normal operation of the mill, when the roll is rotating in the direction indicated by the arrow in FIG. 2 at speeds which are adequate for full hydrodynamic operation, a continuous flow of oil is fed through passageway 29 in the chock, feed openings 30 in the bushing and a rebore 32 in the bearing surface 20. From here, the oil enters between the bearing surface 20 and the rotating journal surface 16 to form a hydrodynamically-maintained oil film 34 at the bearing load zone "Z". The load zone is located on the side opposite to that of the load "L" being applied to the roll.

The oil ultimately escapes axially from between the journal and bearing surfaces 16,18 and is received in inboard and outboard sumps 36,38. From here, the oil is recirculated through filters, cooling devices, etc. (not shown) before being returned to the bearing.

If the rotational speed of the journal surface 16, the load L and the viscosity of the oil all remain within design limits, the bearing will continue to function satisfactorily, with an adequate oil film 34 being hydrodynamically maintained at the load zone Z. However, if one of these parameters should fall below its lower design limit, the hydrodynamically maintained oil film can deteriorate or collapse, causing metal to metal contact between the journal and bearing surfaces 16,20. If this should occur, the resulting friction will rapidly cause bearing failure.

Thus, from zero rotational speed at mill start up to the lower design limit for satisfactory hydrodynamic operation, the oil film 34 at the load zone Z must be created and maintained by means other than the hydrodynamic technique described above. To this end, and with reference additionallY to FIGS. 3–5, in the prior art conventional bearing assemblies, it is known to provide multiple hydrostatic recesses 40 in the bearing surface 20 at the load zone Z. The recesses 40 are interconnected by a network of passageways 42 to a positive displacement, constant volume high pressure oil pump 44.

As viewed radially from inside the bushing, the recesses 40 of the conventional prior art design are generally

rectangular in configuration. The ends of the recesses are feathered as at 46, whereas the bottom 48 and side walls 50 are mutually perpendicular and thus define sharp bottom corners 52. The side walls 50 are perpendicular to the bearing surface 20 to thereby define sharp top edges 53.

With this type of arrangement, as the oil emerges from each recess 40 to hydrostatically form the oil film 34, it encounters very high resistance and thus experiences a significant pressure drop as it is forced in the axial direction between the sharp edges 53 and the bearing surface 20. The net result is that in order to maintain a given oil pressure in the film 34, a substantially higher oil pressure must be maintained in the recess 40. This in turn means that the pump 44 must work harder, and the entire lubrication system must be designed to operate at higher pressures.

It will also be seen that the oil emerging circumferentially from each recess 40 at its feathered ends 46 encountered significantly less resistance as compared to that encountered by the oil emerging axially past the sharp corners 53. This encourages circumferential flow at the expense of axial flow, which in turn adversely affects oil pressure field distribution throughout the load zone. The oil pressure field supports the load at the load zone.

Other disadvantages of the conventional design include high stress concentrations at the bottom corners 52, which can create cracks and cause bearing failure. Also, the sudden change in flow area at the sharp edges 53 results in relatively high fluid velocities, which in turn hasten metal erosion.

The objective of the present invention is to provide a bushing having novel and improved hydrostatic recess configurations which either avoid or at the very least, substantially minimize the problems associated with the prior art.

SUMMARY OF THE INVENTION

According to the present invention, there is provided an improved bushing having hydrostatic recesses which when viewed radially from inside the bushing, have substantially elliptical feathered peripheral edges. The major axes of the recesses extend transversely with respect to the bushing axis, whereas the minor axes of the recesses extend in parallel relationship to the bushing axis. The depth of each recess is non-uniform and increases gradually from its feathered peripheral edge to a maximum at the intersection of the major and minor recess axis.

Each hydrostatic recess of the present invention may be considered as defining a portion of a prolate spheroid produced by the intersection of a double curved surface of revolution with the internal bearing surface of the bushing. The double-curved surface of revolution is typically a torus having an axis of revolution surrounded by the bearing surface of the bushing and extending in parallel relationship to the longitudinal bushing axis.

BRIEF DESCRIPTION OF THE DRAWINGS

An embodiment of the invention will now be described by way of example only with reference to the accompanying drawings, wherein:

FIG. 1 is a longitudinal sectional view taken through an oil film bearing assembly which includes a bushing of conventional design;

FIG. 2 is a cross sectional view on an enlarged scale taken along line 2—2 of FIG. 1;

FIG. 3 is a sectional view taken along line 3—3 of FIG. 2;

FIG. 4 is an enlarged view of a portion of the cross section illustrated in FIG. 2;

FIG. 5 is a cross section taken along line 5—5 of FIG. 4;

FIG. 6 is a view similar to FIG. 3, but on an enlarged scale and showing a bushing according to the present invention;

FIGS. 7 and 8 are sectional views on a greatly enlarged scale taken respectively along lines 7—7 and 8—8 of FIG. 6;

FIGS. 9 and 10 are sectional views taken respectively along lines 9—9 and 10—10 of FIG. 7;

FIG. 11 is a schematic illustration of how the recesses of the present invention are developed at the interior bearing surface of the bushing;

FIG. 12 is an enlarged view taken on line 12—12 of FIG. 11;

FIG. 13 is a view similar to FIG. 4 showing the bushing of the present invention installed in the bearing assembly; and

FIG. 14 is a sectional view taken along line 14—14 of FIG. 13.

DETAILED DESCRIPTION OF ILLUSTRATED EMBODIMENT

Referring now to FIGS. 6–14, a bushing in accordance with the present invention is shown at 60. The bushing has a cylindrical inner bearing surface 62 with a pair of hydrostatic recesses 64 arranged symmetrically on opposite sides of the bearing center.

Each recess 64 is surrounded by a continuous feathered edge 66 which when viewed radially from inside the bushing (as shown in FIG. 6), is substantially elliptical in configuration with a major axis X extending transversally with respect to the longitudinal bushing axis A, and with a minor axis Y which extends in parallel relationship to axis A.

As illustrated in FIGS. 11 and 12, each recess 64 is defined by the intersection of a torus 68 with the internal bearing surface 62. The torus 68 is generated by the revolution of a plane curve 70 about an axis B. The curve 70, which in practice will constitute a cutting edge profile, has a radius R_1. The torus 68 has a radius R_2, with the axis B being located within and parallel to the bushing axis A.

With reference to FIGS. 7 and 8, it will be seen that successive sections taken perpendicular to the major axis X will define segments of a circle each having the radius R_1. Progressing from the center of the recess towards its ends, each such segment has a gradually diminishing depth d and width w. By the same token, and with reference to FIGS. 9 and 10, successive sections taken perpendicular to the minor axis Y will define portions of a circle having the radius R_2. Again progressing from the center of the recess towards its sides, the successive portions have gradually diminishing depths d and lengths 1. It follows, therefore, that each recess 64 defines a portion of a prolate spheroid.

FIGS. 13 and 14 illustrate the bushing of the present invention installed in an oil film bearing assembly. Instead of the sharp cornered side edges of the prior art recesses 40, the continuously feathered peripheral edge 66 of the recess 64 of the present invention accommodates a smoother, and lower velocity flow of axially emerging oil, with a significantly lower accompanying pressure drop. Thus, a lower oil pressure in the recess 64 is required to maintain a given pressure in the hydrostatically maintained film 34. Lower oil velocity reduces erosion of the recess edge 66, and elimination of sharp corners significantly reduces potentially damaging stress concentrations.

Also, the gradually diminishing circumferential flow path resulting from the diminished width w at the ends of the recess (compare FIGS. 7 and 8) resists circumferential flow and further encourages axial flow, thus improving oil distribution throughout the load zone.

I claim:

1. A bushing for use in an oil film bearing assembly of the type employed to rotatably support the journal surface of a rolling mill roll neck, said bushing comprising:

a wall having an inner cylindrical bearing surface adapted to surround said journal surface;

conduit means extending through said wall for conveying liquid lubricant under pressure to a load zone between said bearing surface and said journal surface; and

at least one recess in said bearing surface in communication with said conduit means for receiving the thus conveyed lubricant under pressure and for distributing the same in said load zone in the form of a lubricant film separating said journal surface from said bearing surface, said recess being defined by the intersection of a double-curved surface of revolution with said bearing surface.

2. The bushing of claim 1 wherein said double-curved surface of revolution is a torus.

3. The bushing of claim 2 wherein the rotational axis of said double-curved surface is parallel to the longitudinal axis of said bushing and surrounded by said bearing surface.

4. The bushing of claim 3 wherein the radius of rotation of said double-curved surface of revolution is smaller than the radius of said bearing surface.

5. The bushing of claim 1 wherein said recess defines a portion of a prolate spheroid.

6. The bushing of claim 1 wherein said recess has a substantially elliptical peripheral edge.

7. The bushing of claim 6 wherein said recess has a major axis extending in a direction perpendicular to the longitudinal bushing axis, and a minor axis extending in a direction parallel to the longitudinal bushing axis.

8. The bushing of claim 7 wherein cross sections taken through said recess in directions perpendicular to said major axis define segments of circles.

9. The bushing of either claim 7 or 8 wherein cross sections taken through said recess in directions perpendicular to said minor axis define portions of circles.

10. A bushing for use in an oil film bearing assembly of the type employed to rotatably support the journal surface of a rolling mill roll neck, said bushing comprising:

a wall having an inner cylindrical bearing surface adapted to surround said journal surface;

conduit means extending through said wall for conveying liquid lubricant under pressure to a load zone between said bearing surface and said journal surface; and

a plurality of recesses in said bearing surface in communication with said conduit means for receiving the thus conveyed lubricant and for distributing the same in said load zone in the form of a lubricant

film separating said journal surface from said bearing surface, said recesses being arranged symmetrically with respect to the center of said load zone, each of said recesses having a major axis extending in a direction perpendicular to the longitudinal bushing axis and a minor axis extending in a direction parallel to the longitudinal bushing axis, each of said recesses having a substantially elliptical peripheral edge and a non-uniform depth increasing gradually from said edge to a maximum depth at the intersection of said major and minor axes.

* * * * *

US005516212A

United States Patent [19]

Titcomb

[11] **Patent Number: 5,516,212**

[45] **Date of Patent: May 14, 1996**

[54] **HYDRODYNAMIC BEARING WITH CONTROLLED LUBRICANT PRESSURE DISTRIBUTION**

[75] Inventor: **Forrest D. Titcomb**, Colorado Springs, Colo.

[73] Assignee: **Western Digital Corporation**, Irvine, Calif.

[21] Appl. No.: **529,845**

[22] Filed: **Sep. 18, 1995**

[51] **Int. Cl.**[6] **F16C 37/06**

[52] **U.S. Cl.** **384/107**; 384/112

[58] **Field of Search** 384/100, 107, 384/112, 113

[56] **References Cited**

U.S. PATENT DOCUMENTS

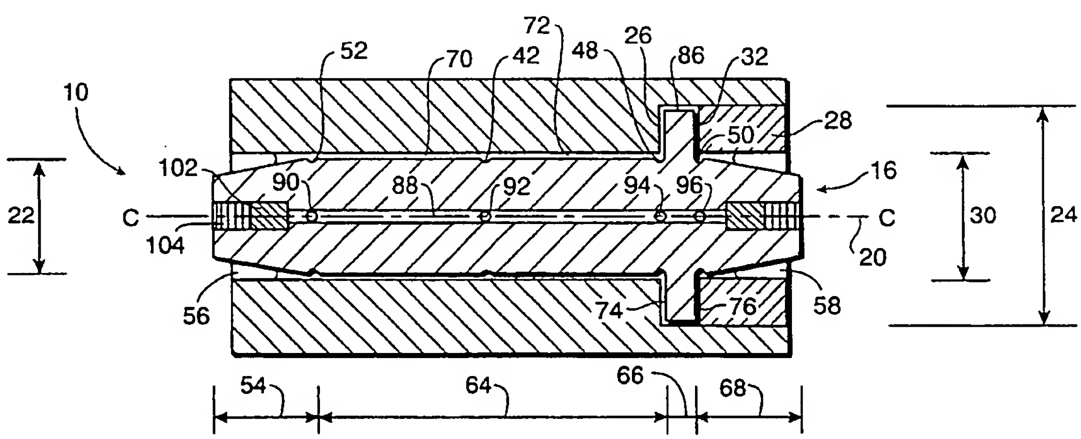
4,795,275	1/1989	Titcomb et al.	384/107
5,067,528	11/1991	Titcomb et al.	141/4
5,112,142	5/1992	Titcomb et al.	384/107
5,246,294	9/1993	Pan	384/119
5,284,391	2/1994	Diel et al.	384/108
5,328,271	7/1994	Titcomb	384/108
5,358,339	10/1994	Konno et al.	384/107
5,407,281	4/1995	Chen	384/107
5,423,612	6/1995	Zang et al.	384/119

Primary Examiner—Thomas R. Hannon
Attorney, Agent, or Firm—James A. Ward; Leo J. Young

[57] **ABSTRACT**

A rotating-shaft hydrodynamic bearing assembly adapted for use in a rotating disk data store. The bearing assembly combines a plurality of spaced-apart radial journal bearings with a two-faced axial thrust plate to provide stiffness against runout at high rotational velocities. Fluid pressure is controlled on both sides of every fluid dement by bounding all radial fluid bearing layers and the dual thrust bearing layers with circumferential undercuts coupled to ambient pressure through a plurality of fluid-filled passages in the rotating shaft. Outward-biased surface-relief patterns are disposed in both axial thrust bearing layers to increase hydrostatic pressure and prevent cavitation in the non-bearing thrust-plate peripheral layer. The fluid bearing layers are disposed in a continuous pressure-controlled fluid film sealed at both ends by surface tension, thereby eliminating sources of air-bubble entrapment. Control of air-bubble entrapment and cavitation eliminates both as sources of surface-tension seal leakage and blowout.

32 Claims, 8 Drawing Sheets



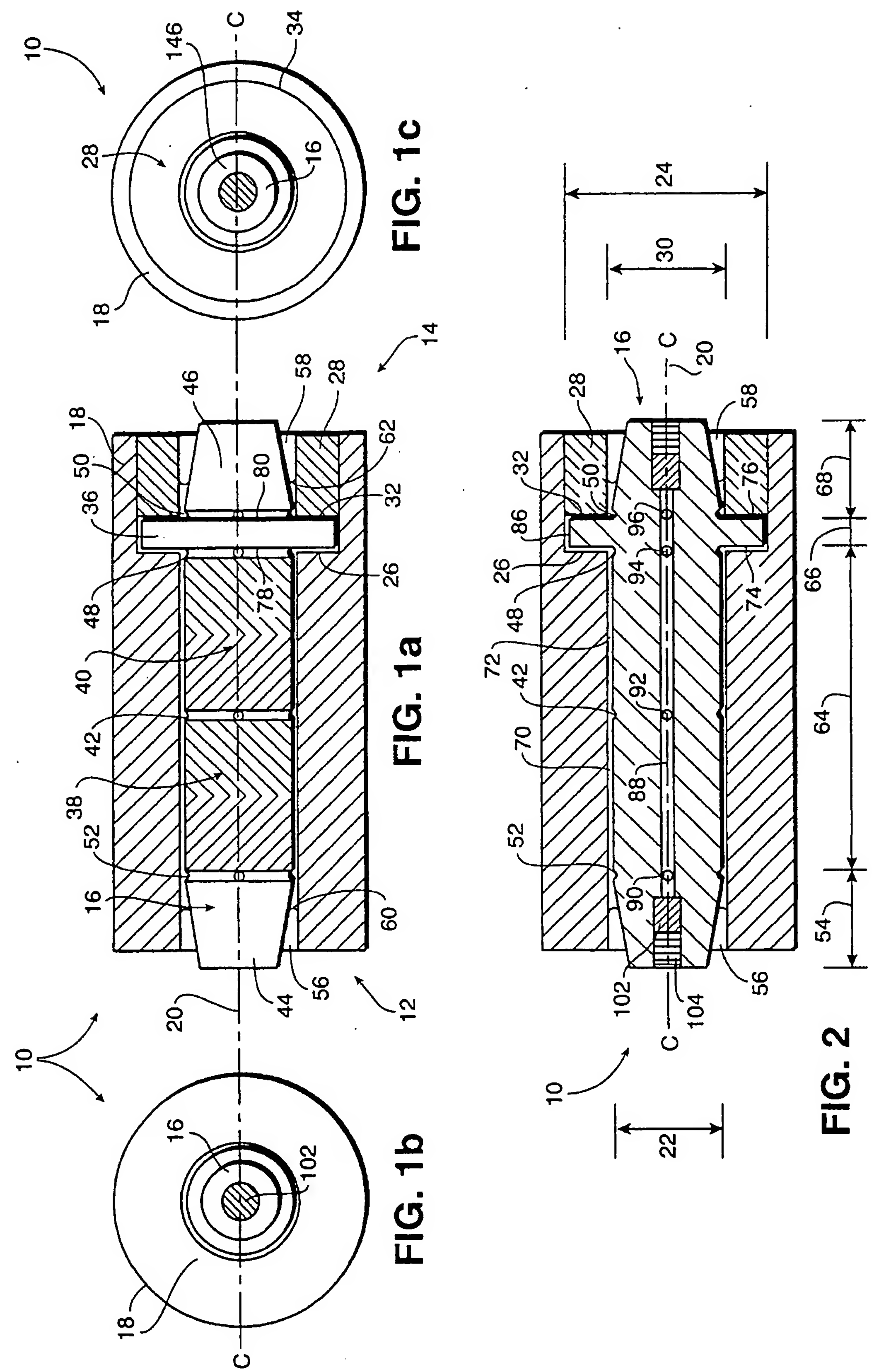


FIG. 1c

FIG. 1a

FIG. 1b

FIG. 2

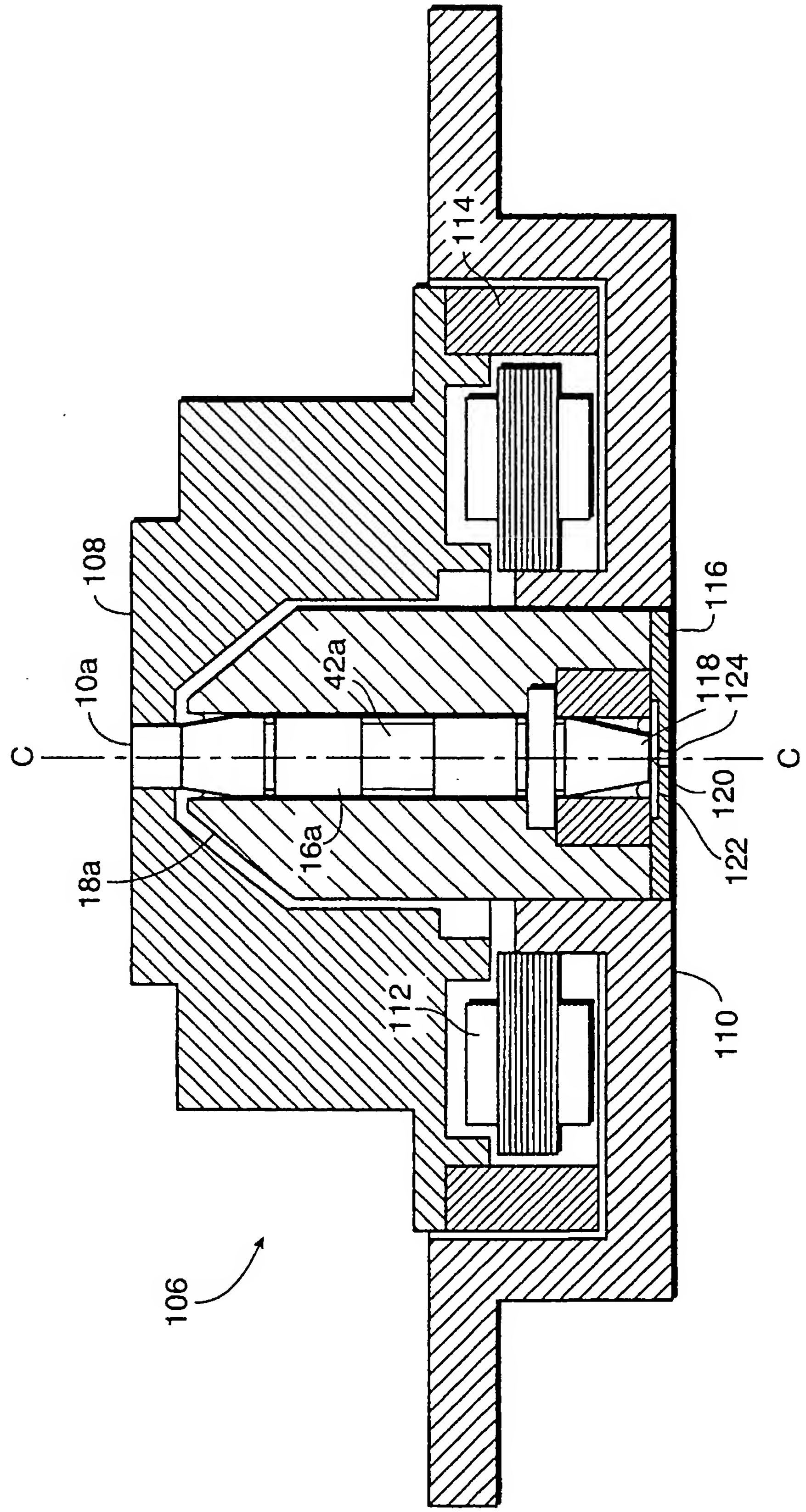
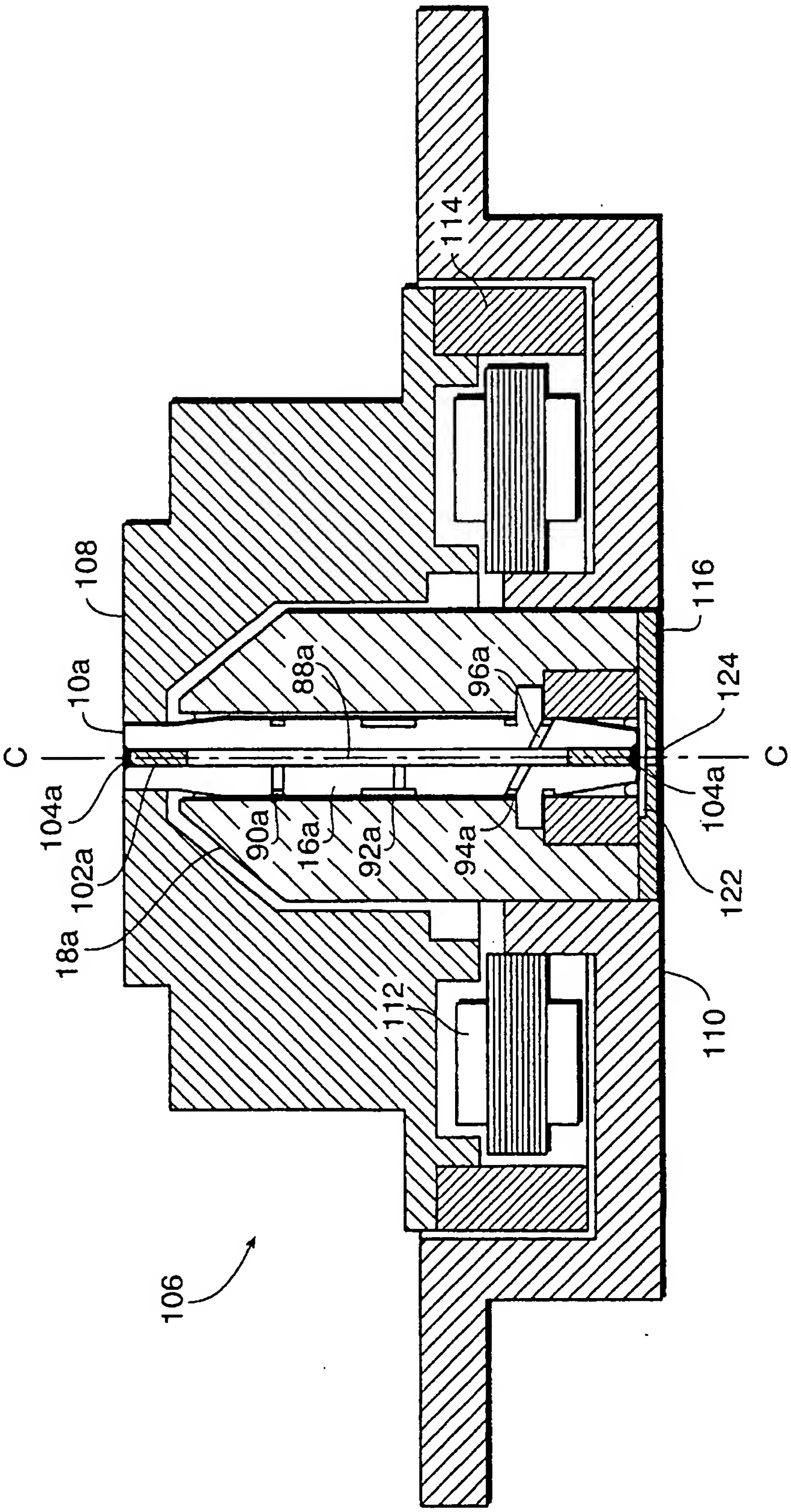


FIG. 3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116
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88a
96a
124
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104a
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102a
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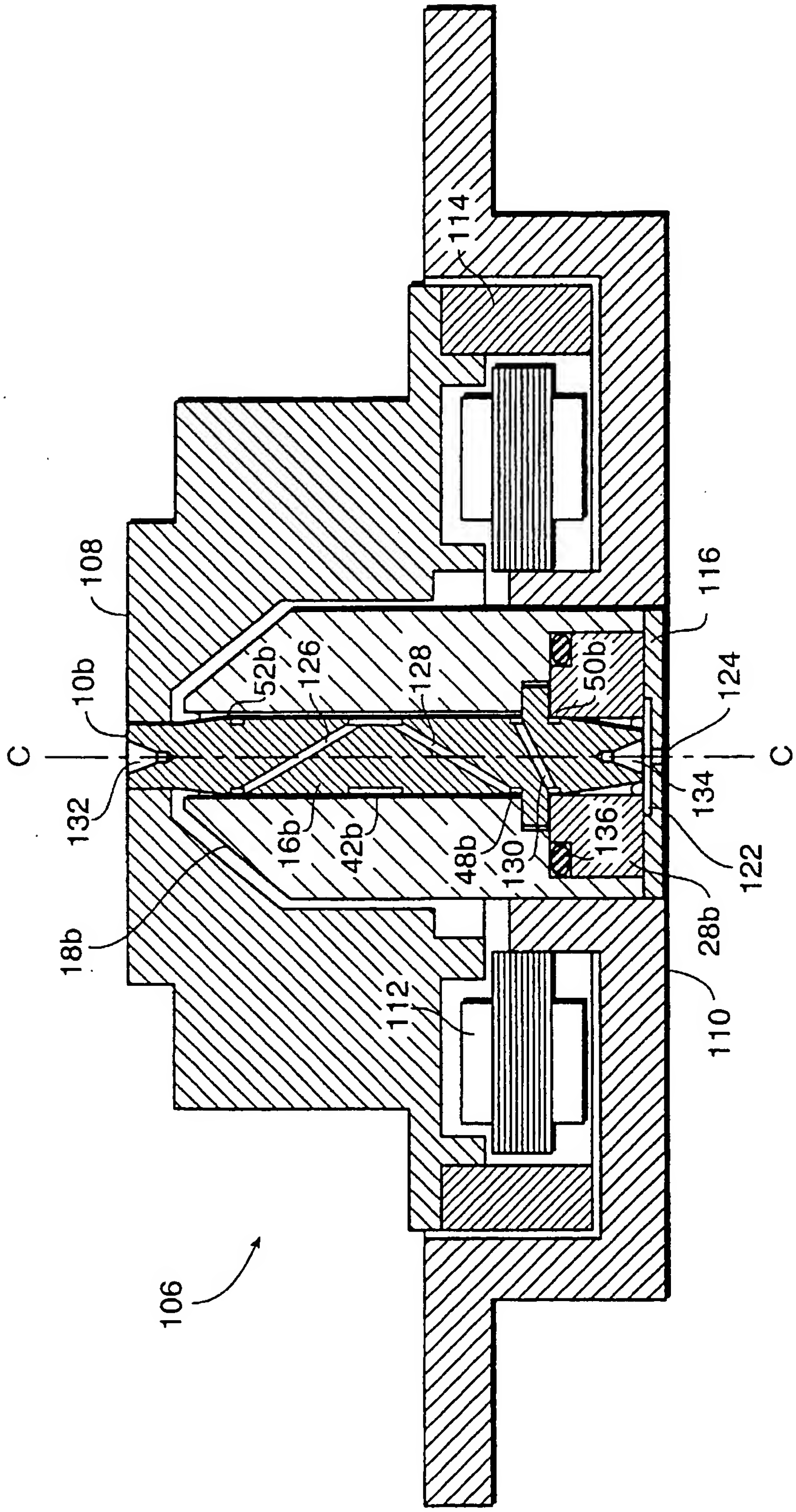
112
110
106

FIG. 4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106

FIG. 5

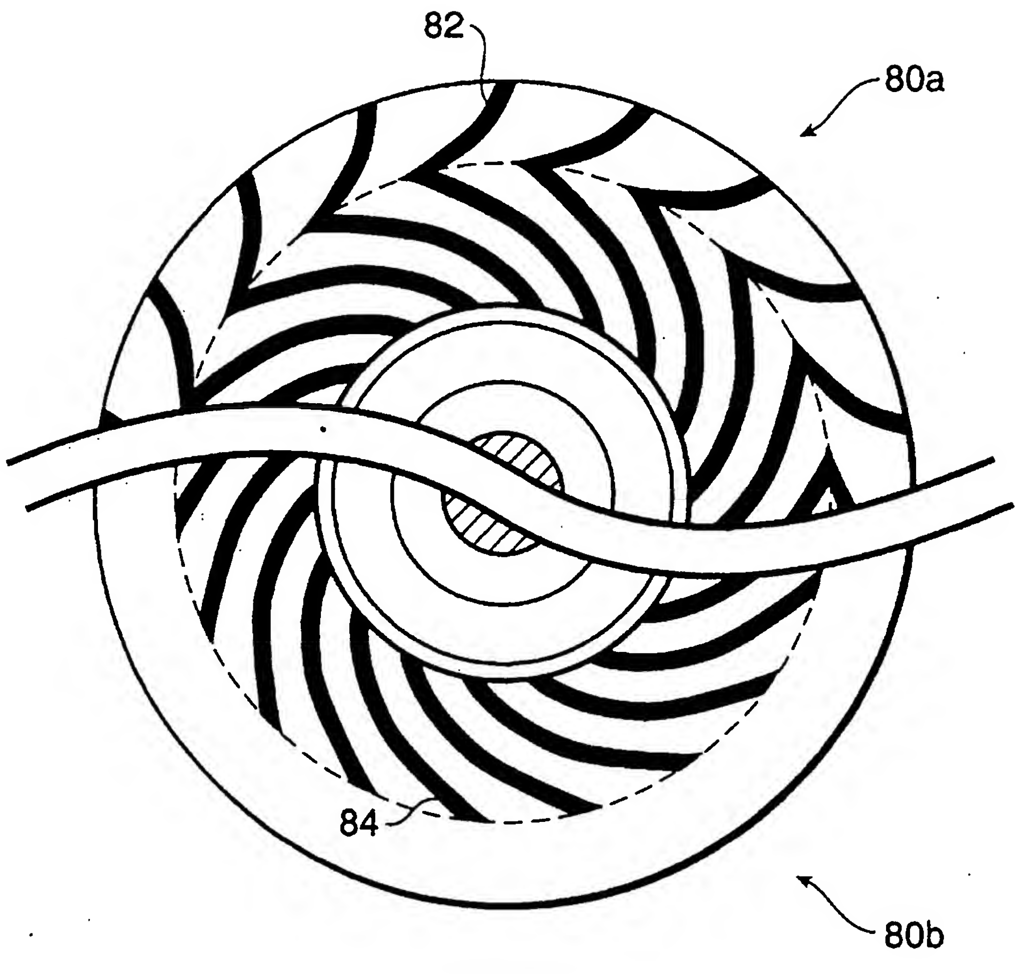
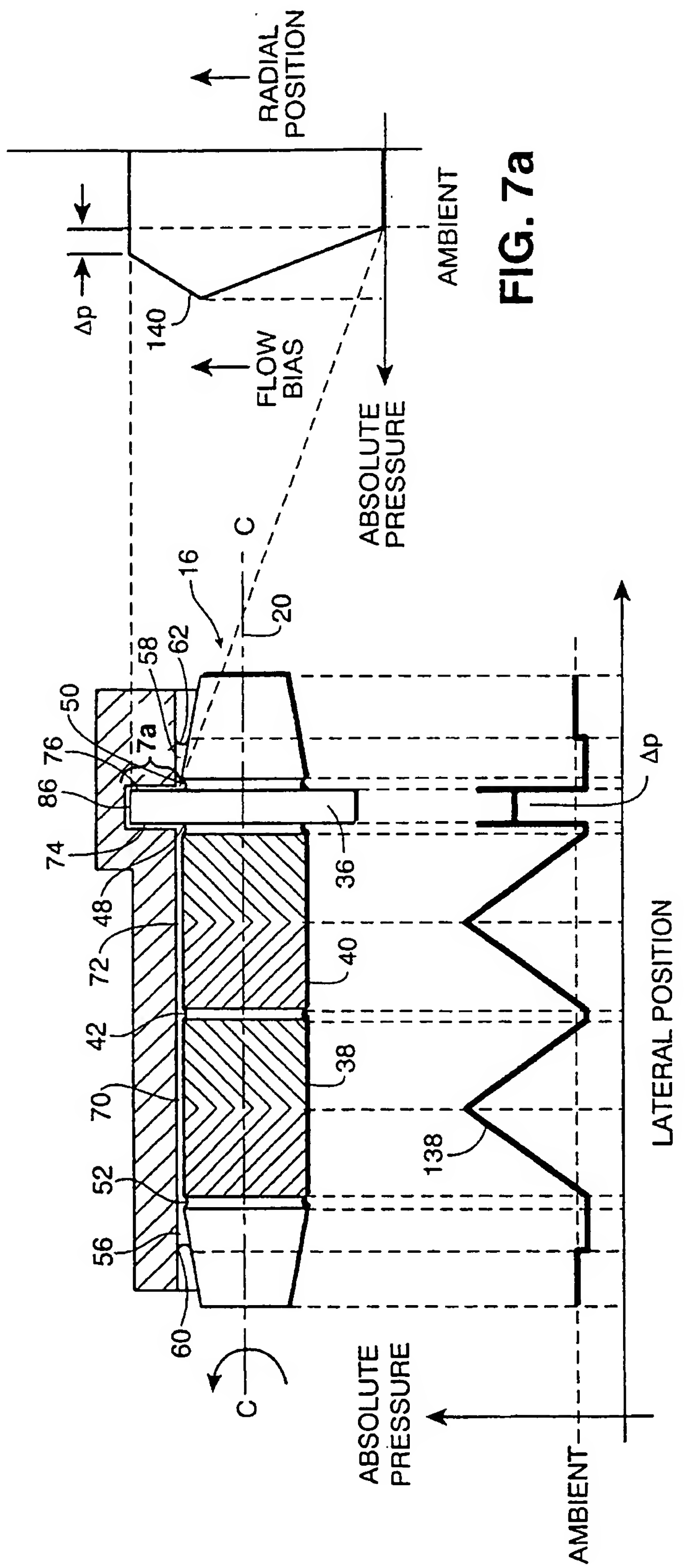


FIG. 6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AMBIENT
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58
50
76
7a
86
74
48
72
42
70
52
56
60
C
36
40
38
138
Δp
LATERAL POSITION
ABSOLUTE PRESSURE
AMBIENT

FIG. 7a

FIG. 7

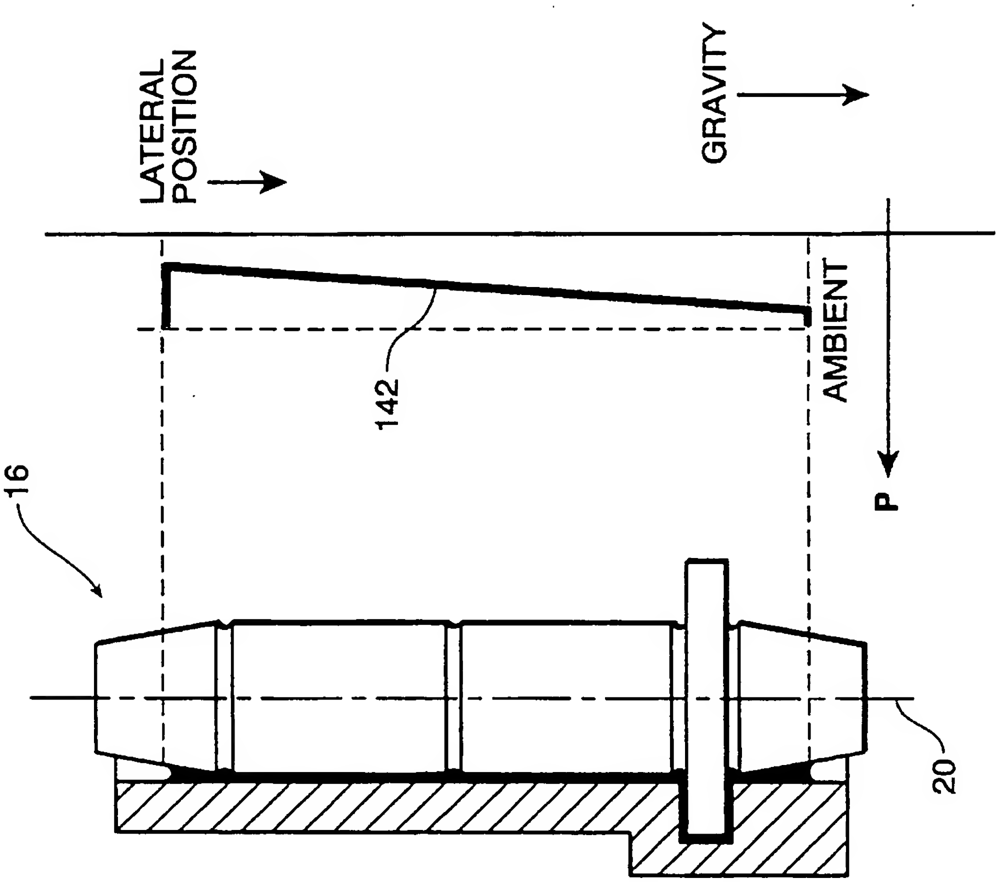
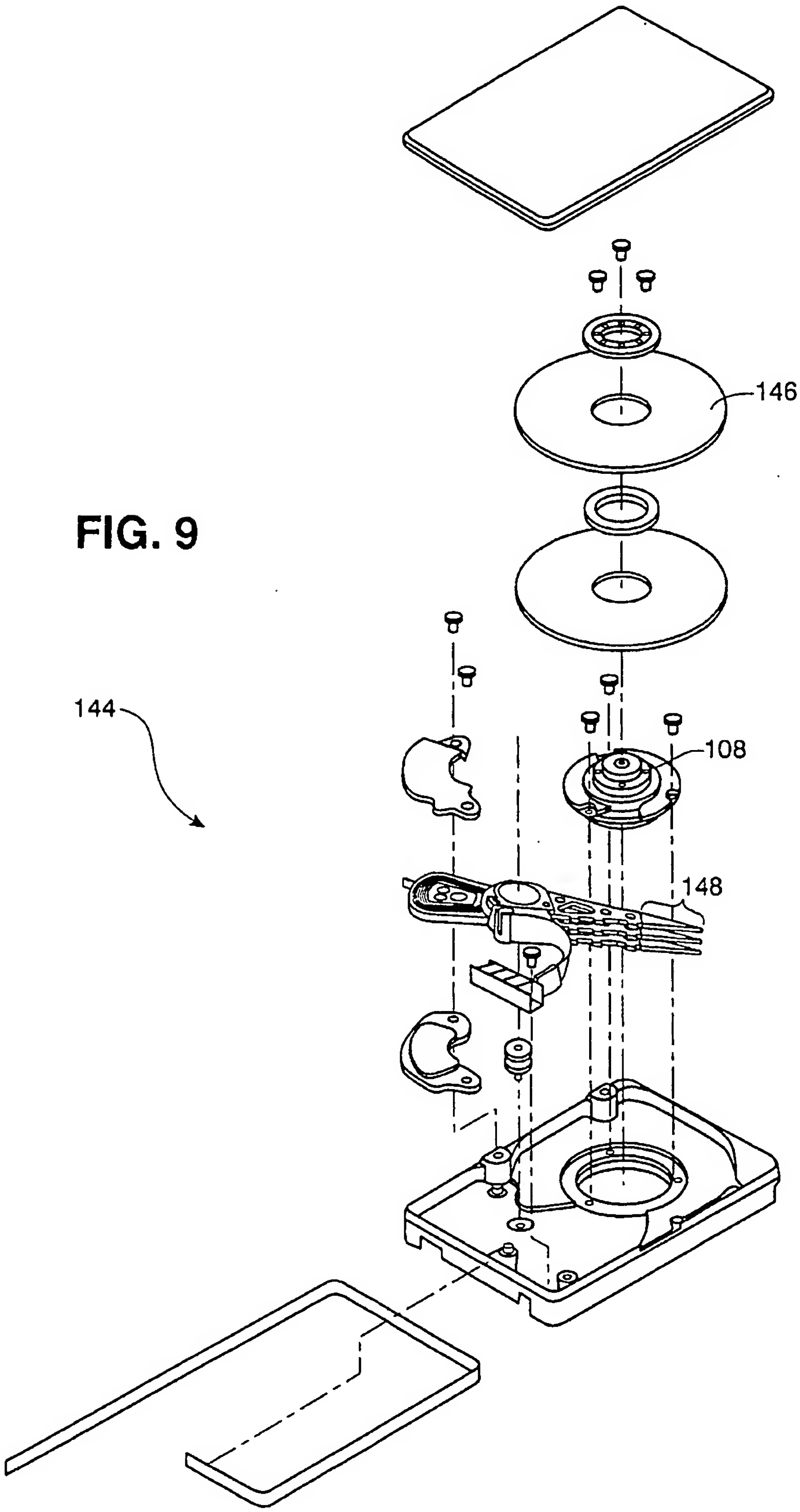


FIG. 8

FIG. 9



HYDRODYNAMIC BEARING WITH CONTROLLED LUBRICANT PRESSURE DISTRIBUTION

BACKGROUND OF THE INVENTION

1. Field of the Invention

This invention relates generally to hydrodynamic bearings for minimizing data storage disk runout at high rotational velocities and specifically to a stiffened rotating-shaft bearing having opposing thrust-bearing axial hydrodynamic pumping action and a plurality of equalized circumferential undercuts for controlling lubricant pressure distribution and eliminating cavitation.

2. Description of the Related Art

Continuing advances in computer data storage technology strongly motivate improvements in magnetic disk areal storage densities. Increased data storage densities require corresponding increases in sensor-to-disk positioning precision. Typically in the art, a sensing head reads or writes streams of data from or to tracks and sectors on the magnetic disk surface. The track width and lineal data density is related to the overall areal storage capacity of the disk surface. Because the typical magnetic disk data store includes several spinning magnetic disks suspended on a common precision spindle bearing assembly, bearing wobble or "runout" directly affects the precise location of microscopic data storage sites on the disk surface with respect to the data sensor.

As the bearing journal and bearing sleeve spin relative to one another, a point on the spin axis may trace out a path or orbit. The wobbling motion of this spin axis includes synchronous and asynchronous components, referred to in the art as repetitive runout and non-repetitive runout, respectively. Hydrodynamic spindle bearing designs are preferred in the disk data store art over the older ball-bearing spindle systems because the rolling elements in ball-bearing spindle systems produce relatively large non-repetitive runout arising from several causes, including imperfect race and ball geometries, surface defects, non-axisymmetric radial stiffness, misalignments and imbalances. Bearing runout limits the practical data storage density, which can be improved only by limiting spindle bearing assembly runout tolerances.

In hydrodynamic bearings, a lubricating fluid (either air or liquid) functions as the actual bearing element between a journal and sleeve in relative rotation. Liquid lubricants such as oil or more complex ferromagnetic fluids are known in the art for use in hydrodynamic bearing assemblies. When a lubricating liquid is used, the liquid itself must be sealed within the bearing assembly to avoid bearing "starvation" arising from fluid loss. Bearing starvation leads directly to increased wear and premature failure of the bearing assembly. In the recent art, such lubricant seals are embodied as "surface-tension" or "capillary" seals at the liquid/air interface of the lubricating fluid. In hydrodynamic bearings using ferromagnetic fluids, the seal may be achieved by establishing a magnetic field at each end of the bearing assembly. Other methods known in the art for avoiding bearing starvation include using centrifugal force to recirculate lubricant flow through passages within the bearing assembly and application of pressure to the bearing surface from an external lubricating fluid source.

The typical hydrodynamic bearing known in the an establishes pumping action in either radial or axial fluid films by defining a series of surface-relief grooves or a single helical surface-relief groove inclined at a particular angle relative to the axis in one of the bearing surfaces. For instance, the journal surface is commonly engraved with a surface-relief pattern disposed to cause the lubricating liquid to be urged toward the center of the journal bearing where it is maintained under pressure for so long as the relative rotation between journal and sleeve is maintained. One type of surface-relief pattern known in the art is the "herringbone" pattern, which may be embodied as a generally symmetrical pattern of repeated Vee-shaped or chevron-shaped relief grooves formed in either the journal or sleeve side of the cylindrical fluid film. The surface on one side of the fluid film is smooth and the relative unidirectional rotation of the grooved and smooth surfaces causes the lubricating liquid to enter the legs of each Vee groove responsive to the urging of liquid flow toward the apex of the Vee. The Vee apex experiences increased fluid pressure arising from the resulting pumping action, thereby creating and maintaining the hydrodynamic bearing layer under a steady pressure generated by the relative rotation of the two surfaces. Usually, the surface-relief patterns are disposed to produce equal and opposite pumping actions so that no net liquid flow occurs in any direction during rotation, thereby minimizing lubricating fluid loss.

The surface-tension or "capillary taper" seal known in the art requires additional measures to balance dynamic fluid pressure distribution within the rotating bearing assembly, such as pressure-equalizing flow passages or external pressure ports. Hydrodynamic bearing assemblies that employ the simple surface-tension taper seal may experience lubricant leakage arising from centrifugal effects of the rotating element (especially in a "fixed shaft" design having a rotating sleeve) and are vulnerable to lubricant blowout arising from entrapped gas bubbles and cavitation. Localized subambient hydrodynamic pressures within the lubricant bearing film may cause cavitation. Even bearing assemblies that use ferromagnetic fluids are known to suffer leakage problems as metallic particles within the ferrosuspension escape over time.

Disadvantageously, many workable lubricant seal designs known in the art require extremely tight clearances and alignments within the hydrodynamic bearing assembly. This often precludes cost-effective manufacture of such assemblies because of rejection or premature failure resulting from even a small deviation or aberration in a component dimension, shape or alignment. Also, as rotational speeds increase, centrifugal forces in the lubricant bearing film increase, thereby increasing stress on the traditional outwardly-tapered capillary seal, eventually causing leakage and lubricant depletion.

The hydrodynamic bearing art is replete with suggestions for improving bearing performance and for reducing fluid leakage during operation. For instance, a useful non-magnetic hydrodynamic bearing design is disclosed by Forrest Titcomb et al. in U.S. Pat. Nos. 4,795,275, 5,067,528, and 5,112,142. Titcomb et al. disclose a rotating shaft and thrust-plate combination disposed within a sleeve to form two pressure equalization ports. They use two thrust plates each with one annular axial thrust bearing layer on the inside face as well as two journal regions providing spaced-apart axial bearing layers to stiffen the bearing assembly against undesired repetitive runout. By adding pressure-equalization passages between the several axial and radial fluid bearing layers, Titcomb et al. prevent pressure buildup and eliminate lubricant fluid flow arising from unequal pressure distributions. Because unequal pressure distributions are eliminated, the two-degree (2°) tapered surface-tension seals at each end of their bearing assembly are sufficient to prevent substantial

fluid leakage, but they do not consider a solution to the effects of bubble-entrapment and cavitation on fluid leakage rates.

In U.S. Pat. Nos. 5,284,391 and 5,328,271, Forrest Titcomb et al. disclose a hydrodynamic bearing assembly that employs a ball-and-socket geometry instead of the journal-and-sleeve geometry commonly known in the art. The "ball" surface is grooved to provide hydrodynamic pumping action in the hemispherical fluid bearing layer, which is sealed by surface-tension at the edges of the hemispherical ball-and-socket clearance. However, Titcomb et al do not consider solutions to the fluid seal and leakage problems related to bubble-entrapment and cavitation.

In U.S. Pat. No. 5,246,291, Coda Pan discloses a flow-regulating hydrodynamic bearing that includes two conical fluid bearing layers instead of the more common combination of orthogonal axial and radial bearing layers. Pan teaches the use of a design having a large reservoir for lubricating oil storage with a covering air volume in communication with ambient pressure and with surface-tension seals formed between respective central passages of shaft and housing end-caps. He provides for recapture of "wandering" lubricant when the bearing assembly is static and relies on centrifugal pumping to throw all statically-trapped oil into the lubricant reservoir during dynamic operation. Pan also proposes attaching a pressure-actuated bladder to the bearing for demand-delivery of lubricant. Thus, Pan resolves the lubricant leakage problem to his satisfaction with a combination of recapture, redelivery and large stand-by lubricant reserves and neither considers nor suggests solutions to the bubble-entrapment and cavitation effects on seal leakage.

In U.S. Pat. No. 5,407,281, Chen discloses a self-replenishing hydrodynamic bearing having a plurality of reservoirs containing a supply of lubricating fluid. He discloses a cyclical herringbone surface-relief pattern that generates alternating localized unidirectional lubricating fluid flow between the reservoirs. By causing fluid flow to alternate between adjacent grooves, Chen manages to provide localized flushing flow without incurring a net fluid flow sufficient to blow out the surface-tension seals. However, he neither considers nor suggests solutions to the bubble-entrapment and cavitation problems that may cause seal leakage or blowout.

In U.S. Pat. No. 5,358,339, Konno et al. disclose a hydrodynamic bearing assembly that employs liquid radial-bearing layers and gaseous axial-bearing layers in the same assembly. By eliminating liquid in the annular axial thrust bearing layer, they avoid fluid scattering arising from centrifugal forces, which eliminates one source of lubricating fluid leakage known in the art. Konno et al. also disclose a "chamfer" geometry for providing a surface-tension seal at the radial-bearing boundaries. However, Konno et al. neither consider nor suggest solutions to the bubble-entrapment and cavitation problems that may aggravate surface-tension seal leakage.

In U.S. Pat. No. 5,423,612, Yan Zang et al. disclose a hydrodynamic bearing and seal that includes a plurality of spaced-apart radial journal bearings and a single annular axial thrust plate with two surfaces each defining one side of two hydrodynamic thrust bearings. They use spaced-apart radial bearings to improve shaft stiffness for less repetitive runout in the bearing assembly. Although Zang et al. use surface-relief pumping patterns on both faces of their thrust plate, they rely on centrifugal force alone to hold the lubricating fluid within both axial bearing layers during rotation and neither consider nor suggest means to balance flow and avoid cavitation in the nonbearing fluid layer at the thrust plate periphery. Moreover, although Zang et al. use two radial bearing layers that are separated by substantial distance to improve stiffness, they do not suggest solutions to the bubble-entrapment problem.

Without a reasonable solution to the capillary seal leakage problems caused by air-entrapment and cavitation, practitioners in the art are obliged to either accept large runout in less stiff bearings or perhaps to resort to large fluid reservoirs and costly tight-tolerance clearance specifications to avoid unacceptable reduction in bearing life expectancy caused by premature lubricating fluid loss. These unresolved problems and deficiencies are clearly felt in the art and are solved by this invention in the manner described below.

SUMMARY OF THE INVENTION

This invention solves the above problems by combining several elements to control lubricating fluid pressure distribution in a spinning-shaft hydrodynamic bearing having two or more radial and at least two axial thrust bearing layers. These include stiffening the bearing assembly by separating the two radial bearing layers and adding circumferential undercuts in either the shaft or sleeve on each side of both radial bearing clearances and on each side of the thrust plate that supplies the two axial bearing layers. Surface-relief patterns are incorporated on both sides of the thrust plate to urge radially-outward fluid flow in both axial bearing clearances, which prevents cavitation by raising the hydrostatic pressure in the non-bearing clearance at the thrust plate periphery. All circumferential undercuts are interconnected by pressure-equalizing passages in the shaft, which may include a single axial passage interconnecting various radial and/or oblique passages. The fluid bearing layers are thus disposed to form a continuous pressure-controlled fluid film sealed at each end by diverging tapered surface-tension seals formed in tapered clearances between shaft and sleeve, thereby eliminating the usual sources of air-bubble entrapment. This control of air-bubble entrapment and cavitation eliminates both as sources of surface-tension seal leakage and blowout.

One object of this invention is to ensure balanced hydrostatic pressure throughout the entire bearing assembly in all of the several fluid bearing layers. It is a feature of this invention that both radial fluid bearing layers are disposed between circumferential undercuts, either in the shaft or in the sleeve. It is yet another feature of this invention that fluid pressure is controlled on both sides of every bearing element by coupling all radial fluid bearing layers and axial thrust bearing layers to a circumferential undercut (or overcut) and by coupling these circumferential undercuts to ambient pressure through a plurality of fluid-filled passages in the rotating shaft.

It is another object of this invention to prevent cavitation by eliminating any possibility of localized subambient hydrostatic pressure such as may occur in the non-bearing clearance between the thrust-plate periphery and the sleeve. It is a feature of this invention that the axial fluid bearing layers on each side of the thrust plate are urged radially-outward in balance to produce elevated static pressure at the thrust plate periphery without net fluid flow.

In a further refinement of this invention, a barrier film is provided on the surfaces bounding the tapered clearances at each end of the bearing, thereby discouraging lubricating fluid migration from the surface-tension seals at each end.

It is an advantage of the hydrodynamic bearing of this invention that it is adaptable for use in a rotating disk data storage device of the type commonly employed with host computers. It is a feature of this invention that one end of the shaft can be extended beyond the sleeve to permit mounting of a disk thereon.

It is an advantage of this invention that it permits scaleability and ease of manufacture. It is another advantage of this invention that it provides improved axial and radial stiffness, thereby reducing repetitive runout. It is yet another advantage of this invention that it requires relatively few components.

The foregoing, together with other objects, features and advantages of this invention, can be better appreciated with reference to the following specification, claims, and the accompanying drawing.

BRIEF DESCRIPTION OF THE DRAWING

For a more complete understanding of this invention, reference is now made to the following detailed description of the embodiments as illustrated in the accompanying drawing, in which like reference designations represent like features, and wherein:

FIG. 1*a*–1*c* show a partial cross-sectional side view and both end views of a first embodiment of the hydrodynamic bearing of this invention;

FIG. 2 shows a full cross-sectional side view of the hydrodynamic bearing from FIGS. 1*a*–1*c;*

FIG. 3 shows a partial cross-sectional side view of a hard-disk drive motor assembly employing a second embodiment of the hydrodynamic bearing of this invention;

FIG. 4 shows a full cross-sectional side view of the hard-disk drive motor assembly from FIG. 3;

FIG. 5 shows a full cross-sectional side view of a third embodiment of the hard-disk drive motor assembly of this invention;

FIG. 6 shows an end view of the bearing shaft element of the hydrodynamic bearing assembly from FIGS. 1*a*–1*c* with two exemplary thrust-plate face surface-relief patterns suitable for use in this invention;

FIG. 7, including detail FIG. 7*a*, is a schematic diagram illustrating the hydrostatic pressure distribution at full rotational velocity along the radial and axial fluid bearing clearances in the bearing assembly of this invention;

FIG. 8 is a schematic diagram illustrating the hydrostatic pressure bias arising from vertical disposition of the hydrodynamic bearing assembly of this invention; and

FIG. 9 is an exploded perspective diagram of an illustrative hard-disk data store apparatus employing the hydrodynamic bearing assembly of this invention.

DETAILED DESCRIPTION OF THE PREFERRED EMBODIMENTS

FIGS. 1*a*–1*c* show an exemplary embodiment of the hydrodynamic bearing assembly **10** of this invention, which has a first end **12**, a view of which is shown in FIG. 1B and second end **14**, a view of which is shown in FIG. 1*c*. FIG. 1*a* shows hydrodynamic bearing assembly **10** from the side. In a partial cross-section revealing the surface of a bearing shaft **16** disposed within a cross-sectional view of the substantially cylindrical support sleeve **18**.

FIG. 2 shows a complete cross-sectional side view of hydrodynamic bearing assembly **10**, which is disposed symmetrically about a bearing axis of rotation **20**. Support sleeve **18** is shown having a first inner diameter **22** at first assembly end **12**. Inner diameter **22** is stepped up to a second sleeve inner diameter **24** at second assembly end **14**, thereby forming a sleeve shoulder **26**. The effective inner diameter of support sleeve **18** is stepped down again at assembly end **14** by the insertion of a ring-shaped plug **28** having the inner diameter **30**. One wall of plug **28** forms a second sleeve shoulder **32** when inserted fixedly as shown. Although inner sleeve diameter **22** and inner plug diameter **30** are shown as identical for illustrative purposes, an important advantage of the hydrodynamic bearing assembly **10** of this invention is that inner diameters **22** and **30** may differ, if desired, without compromising the effectiveness of the surface-tension seals formed at bearing ends **12** and **14**.

Bearing shaft **16** is inserted into support sleeve **18** and disposed rotatably therein before inserting ring-shaped plug **28**, which is secured to support sleeve **18** at the margin **34** by any useful means known in the art, such as epoxy adhesive, shrink-fitting or the like. Bearing shaft **16** includes a thrust plate **36** disposed adjacent to the two journals **38** and **40**. Although journals **38** and **40** are shown as immediately adjacent to one another on bearing shaft **16**, they may be instead disposed separated by a wider undercut (not shown), thereby increasing the stiffness of bearing assembly **10**.

Thrust plate **36** and journals **38** and **40** are disposed between two tapered shaft ends **44** and **46**. Thrust plate **36** is disposed between two circumferential undercuts **48** and **50**, which should be created by reducing the outer diameter of bearing shaft **16** substantially as shown. Although undercuts **48** and **50** are shown as having a semicircular profile, any undercut profile is suitable so long as the depth and width satisfy the important fluid pressure equalization design requirements of this invention. Similarly, each of the two journals **38** and **40** are disposed between two circumferential undercuts. Journal **40** is disposed between undercut **48** and intermediate undercut **42**. Journal **38** is disposed between intermediate undercut **42** and the circumferential undercut **52**.

Although circumferential undercuts **42, 48, 50** and **52** are shown in FIGS. 1–2 as regions of reduced outer bearing shaft diameter, only undercuts **48** and **50** are preferably disposed in bearing shaft **16**. All other undercuts may equally well be embodied as regions of increased inner diameter of similar volume in support sleeve **18**. That is, intermediate undercut **42** and undercut **52** can equally well be embodied as circumferential overcuts (not shown) in support sleeve **18**. As mentioned above, the exact volume, depth and profile of the undercut or overcut is an important design consideration for the hydrodynamic bearing assembly of this invention because the undercut fluid volume determines the reserve lubricating fluid availability to the fluid films in the bearing clearances, as described in more detail hereinbelow.

Bearing shaft **16** is disposed rotatably within support sleeve **18** to form a plurality of bearing and non-bearing clearances, which are now described. The outer surface of tapered shaft end **44** and the inner surface of the untapered sleeve end **54** define a first tapered clearance **56**. Similarly, the outer surface of tapered shaft end **46** and the untapered inner diameter of ring-shaped plug **28** define a second tapered clearance **58**. Tapered clearances **56** and **58** are disposed to create surface-tension seals that retain the lubricating fluid within the interconnected bearing clearances. Tapered clearance **56** creates the surface-tension seal **60** and

tapered clearance 58 creates the surface-tension seal 62. Surface-tension seals 60 and 62 each consist of a substantially annular meniscus serving to seal the lubricating fluid within the bearing clearances. The degree of taper shown in the drawing has been exaggerated for clarity. In practice, a taper angle of less than five degrees (5°) is preferred. Similarly, the clearance dimensions defined by the respective sleeve and shaft surfaces and the undercut dimensions are also exaggerated in the drawing for purposes of clarity and may in practice be three orders of magnitude less than support shaft inner diameter 22.

Support sleeve 18 includes a radial bearing zone 64 and axial thrust bearing zone 66 disposed between the two untapered sleeve ends 54 and 68. The two journals 38 and 40 cooperate with inner support sleeve diameter 22 to form two roughly cylindrical radial-bearing clearances 70 and 72, respectively. That is, journal 38 defines cylindrical radial bearing clearance 70 and journal 40 defines cylindrical radial bearing clearance 72. Clearances 70 and 72 are filled with the lubricating fluid that also fills undercuts 42, 48 and 52 and all other clearances between seal 60 and seal 62.

In FIG. 1, journals 38 and 40 are shown with a herringbone surface-relief pattern disposed to increase hydrodynamic pressure within clearances 70 and 72 responsive to shaft rotation. These herringbone patterns may also be etched into radial bearing zone 64 of support sleeve 18 to equal effect. Hydrodynamic pumping occurs responsive to relative motion between a smooth surface and a very closely-disposed surface having the herringbone surface-relief pattern, whether or not the pattern is on the stationary surface or the moving surface. During rotation the precise separation between the shaft surface and sleeve surface at any particular locus depends on the applied radial load and the hydrodynamic pressure. When the local separation is reduced by lateral displacement, the hydrodynamic pumping action responsively increases local hydrodynamic pressure, thereby applying a force tending to increase separation. Thus, the hydrodynamic pumping action of the herringbone surface-relief pattern on journals 38 and 40 tends to force both clearances 68 and 70 into substantial cylindrical symmetry. For this reason, these clearances are herein denominated as "substantially cylindrical".

Note that radial bearing zone 64 includes two spaced-apart radial bearing clearances 70 and 72. This represents an important feature of this invention. Although journals 38 and 40 are shown separated by a relatively narrow undercut 42, separating clearances 70 and 72 on axis 20 by a wider distance is preferred to obtain advantageous stiffening of assembly 10 that reduces runout arising from temporary asymmetries within clearances 70 and 72.

Thrust plate 36 on shaft 16 cooperates with axial thrust bearing zone 66 in sleeve 18 to form two annular thrust-bearing clearances 74 and 76. Annular thrust-bearing clearance 74 is formed between sleeve shoulder 26 and the axis-normal plate face 78 on one side of thrust plate 36. Similarly, annular thrust bearing clearance 76 is formed between the axis-normal plate face 80 on the other side thrust plate 36 and second sleeve shoulder 32.

FIG. 6 provides a schematic illustration of plate face 80 exemplifying the surface-relief pattern used thereon. The patterns on plate faces 78 and 80 differ only in that they are mirror images of one another. Face 80*a* in FIG. 6 is shown with exemplary surface-relief pattern 82 and face 80*b* is shown with exemplary pattern 84. Either pattern 82 or 84 cooperates with a smooth surface on the respective shoulders 26 and 32 to provide hydrodynamic pumping of the lubricating fluid in a radially-outward direction within annular clearances 74 and 76. Of course, surface-relief pattern 82 (or 84) and its mirror image may also be disposed on the respective shoulders 32 and/or 26, in which case the matching surface of plate faces 78 and/or 80 would be smooth. The hydrodynamic pumping effect within clearances 74 and 76 arises from the relative motion of two annular surfaces, one of which has a surface-relief pattern such as pattern 82 or 84 and the other of which smooth. It is an important element of this invention that both clearances 74 and 76 operate in balance to urge the lubricating fluid in a radially-outward direction, thereby increasing static hydrodynamic pressure without net flow within the substantially-cylindrical non-bearing clearance 86 at the periphery of thrust plate 36. Another important element of this invention is the location of circumferential undercuts 48 and 50 in shaft 16 immediately on each side of thrust plate 36. Undercuts 48 and 50 provide reserve lubricating fluid, couple the axial bearing layer pressure to ambient, connect annular clearance 74 to cylindrical clearance 72 and connect annular clearance 76 to tapered clearance 58.

It may now be appreciated, with reference to the above description, that the interconnected fluid bearing layers within bearing assembly 10 are fully coupled from tapered clearance 56 through circumferential undercut 52 to radial bearing layer 70 and therefrom through cylindrical undercut 42 to radial bearing layer 72 and therefrom through cylindrical undercut 48 to axial thrust bearing layer 74 and therefrom through non-bearing clearance 86 to axial thrust-bearing layer 76 and therefrom through cylindrical undercut 50 to tapered clearance 58. The elevated hydrostatic pressure in non-bearing clearance 86 operates to prevent the cavitation that can occur in that region because of negative hydrostatic pressures fluctuations caused by interaction of thrust-bearing layers 74 and 76. The coupling of the axial bearing layers 74 and 76 seamlessly to radial bearing layers 70 and 72 for the first time eliminates gas-bubble entrapment at the multiple capillary seals normally used in the art. The entire lubricating fluid volume within bearing assembly 10 is sealed only on each end by surface-tension seals 60 and 62.

FIG. 2 provides a cross-sectional representation of bearing shaft 16 that shows a plurality of internal passages disposed to equalize hydrodynamic pressure throughout assembly 10. That is, this invention provides direct pressure equalization between all circumferential undercuts 42, 48, 50 and 52 by coupling them to one another through a plurality of passages. Preferably, a plurality of radial or oblique passages are coupled centrally by a central axial passage 88 extending from one end to the other of shaft 16. Radial passages 90, 92, 94 and 96 connect central passage 88 with circumferential undercuts 52, 42, 48 and 50, respectively. Passages 88, 90, 92, 94 and 96 also provide an additional lubricating fluid reservoir to dampen fluctuations in local pressure. Fabrication of axial passage 88 in shaft 16 can be accomplished by machining and insertion of a plug 102 and appropriate seal 104 to cap passage 88 at both ends of shaft 16, substantially as shown.

The lubricating fluid within assembly 10 may be a shearing oil such as a polyalphaolefin oil as known to practitioners skilled in the art. For instance, it has been found that NYE132B or NYE179 oil from the W. F. Nye Corp., Bedford, Mass. is suitable for use with the bearing assembly of this invention.

The exact dispositions of the surface-tension seal menisci 60 and 62 depend on the degree of taper in tapered clearances 56 and 58, respectively, and also depend on the

"wetting" characteristics of the shaft and sleeve surfaces that form clearances **56** and **58**. The axial surface-tension forces forming capillary seals **60** and **62** depend on the length of the wetted perimeter of the liquid-gas interface, the liquid lubricant surface-tension, the taper angle and the contact angle. The axial positioning of menisci **60** and **62** varies with internal hydrodynamic pressure arising from bearing operation, first moving at startup and then stabilizing when the surface-tension forces and the internal hydrodynamic pressure forces balance. Sudden large increases in internal hydrodynamic pressure that can result from gas-bubble entrapment or cavitation may cause seal blow-out or fluid leakage. This invention eliminates these sources of surface-tension seal failure by using an opposing thrust bearing patterning scheme and by controlling hydrodynamic pressure throughout the interconnected plurality fluid-filled clearances, as discussed above. The actual hydrostatic pressure distribution of this invention within the lubricating fluid layers is described below in connection with FIGS. **7–8**.

It has been found that applying a barrier-film coating on each pair of surfaces forming tapered clearances **56** and **58** prevents migration of the lubricating fluid from menisci **60** and **62**. For instance, a coating of NYEBAR (a trademark of William F. Nye Corp, supra) prevents wetting of the surfaces by the lubricating fluid, thereby increasing the meniscus contact angles sufficiently to eliminate fluid migration. As is known for normal uncoated metal surfaces, a lubricating oil migrates along the surface by wetting to create a meniscus angle of about zero degrees. Using a barrier-film coating increases the meniscus contact angle to about 75 degrees (for NYEBAR), thereby eliminating most migration and spontaneous surface wetting in the seal regions.

Bearing assembly **10** is adapted for use in a data store apparatus as part of the disk transport mechanism **106** shown in FIG. **3**, which presents a partial cross-sectional view of mechanism **106**. Assembly **10a** is shown disposed vertically with the thrust plate down. Shaft undercut **42a** is shown having a width that is preferably larger than the undercut width shown in FIGS. **1a–1c** and **2**. The shaft **16a** is captured in a sleeve **18a** having an external shape and dimensions tailored to disk transport mechanism **106**. A mounting hub **108** is affixed to the upper end of shaft **16a** for rotation therewith. Mechanism **106** has a base **110** adapted for mounting within the data store apparatus (not shown). A stator **112** and a ting magnet **114** form a motor to turn mounting hub **108** (and shaft **16a**) at high speed. A dust cover **116** protects the end base member **118** and end face **120** of shaft **16a**. A clearance **122** is provided between end face **120** and dust cover **116**, wherein an optional grounding brush (not shown) may be situated. Dust cover **116** includes a porous port **124** to provide the necessary atmospheric pressure venting for the liquid-gas interface of the surface-tension seals discussed above. The enlarged circumferential undercut **42a** shows that the undercut and patterning features of this invention, may vary in size and placement depending upon the particular implementation. For example, in a hard disk drive for a notebook computer, the entire length of assembly **10a** may be less than 2.5 centimeters and yet support a variety of disk platter diameters and weights or even a plurality of stacked disk platters (not shown).

FIG. **4** shows the disk transport mechanism **106** from FIG. **3** in full cross-section, thereby revealing passages **88a**, **90a**, **92a**, **94a** and **96a** within shaft **16a**. Note that passages **94a** and **96a** are disposed obliquely instead of radially, thereby demonstrating that the precise passage geometry used in this invention may be varied for ease of construction or for other purposes. For instance, the inventor has found that a single oblique drill hole can be used to fabricate both passages **94a** and **96a**, thereby simplifying manufacture without affecting the hydrodynamic pressure equalization control of this invention. Central axial passage **88a** is sealed at each end by a plug **102a** and a weld **104a** tailored to the adaptation for disk transport mechanism **106**.

FIG. **5** shows an alternative embodiment contemplated by the inventor for use of the hydrodynamic bearing assembly of this invention. Bearing assembly **10b** includes a bearing shaft **16b** with the oblique passages **126**, **128** and **130**. Oblique passage **126** connects undercuts **52b** and **42b**. Oblique passage **128** connects undercuts **42b** and **48b**. Passage **130** connects undercuts **48b** and **50b**. Accordingly, because undercuts **52b**, **42b**, **48b** and **50b** are interconnected using only the three oblique passage **126**, **128** and **130**, no central axial passage is necessary in shaft **16b**. Elimination of the central axial passage also eliminates all plugs and seals. In this embodiment, the conical cavities **132** and **134** are retained in the ends of shaft **16b** to eliminate the additional machining that otherwise would be necessary to flatten and seal the ends. Ring-shaped plug **28b** is shown using an O-ring seal **136**.

FIG. **7** is a schematic diagram of the hydrostatic lubricating fluid pressure distribution along the fluid-filled bearing clearances described above. The pressure profile **138** shows fluid pressure as a function of lateral position along axis **20**. The pressure profile **140** shows hydrostatic fluid pressure as a function of radial position and is substantially identical for both axis-normal faces **74** and **76** on thrust plate **36**. Beginning at meniscus **60**, the pressure drops slightly from ambient to accommodate the surface-tension loss at the air/fluid interface. This pressure remains constant over tapered clearance **56** and within circumferential undercut **52**. The pressure rises sharply along radial bearing clearance **70** because of the hydrodynamic pumping action of the surface-relief pattern on journal **38**. The symmetric herringbone pattern on the surface of journal **38** causes the pressure profile to peak at the middle and fall back symmetrically to the slightly less than ambient level at circumferential undercut **42**, where it remains across the entire width of undercut **42**. The pressure profile in the second radial bearing clearance **72** is substantially the same as that shown for clearance **70** because the pattern on journal **40** is substantially the same as that for journal **38**. The pressure within undercut **48** remains slightly less than ambient because it is coupled to undercuts **42**, **52** and **50** by the journal passages discussed above in connection with FIGS. **2** and **4–5**. Hydrostatic pressure within non-bearing clearance **86** at the periphery of thrust plate **36** is elevated by Δp (see FIG. **7a**) above ambient because of the opposing outwardly-biased pumping actions of axial bearing clearances **74** and **76**. Again, the pressure returns to slightly below ambient at undercut **50** and remains there within and across tapered clearance **58** until it returns to ambient at meniscus **62**.

FIG. **7a** shows the hydrostatic pressure distribution along a radius of axial bearing clearance **76**, which is representative of the radial pressure distribution over both annular thrust bearing clearances **74** and **76**. As mentioned above in connection with FIGS. **1–2** and **6**, the surface-relief patterns on faces **78** and **80** are both disposed asymmetrically to urge axially-outward fluid flow, causing the pressure increases with radial position until it peaks at a radius more than half-way to the thrust plate periphery, at which point it begins to fall symmetrically toward the reduced pressure level (Δp) within non-bearing clearance **86**. This reduced "thrust-periphery" pressure Δp prevents cavitation within clearance **86**. Both axial bearing clearances **74** and **76**

enclose "outward-pumping" bearing layers that force an increase in fluid pressure within clearance **86** without net fluid flow around thrust plate **36**.

It can be readily appreciated with reference to the above discussion in connection with FIGS. **7** and **7***a* that the continuous sealed fluid layer from meniscus **60** to meniscus **62** offers no opportunity for entrapping gas bubbles during rotation of shaft **16**. Moreover, examination of both pressure profiles **138** and **140** shows conclusively that, except for surface tension losses, there is no localized negative pressure and thus no possible fluid cavitation during operation.

FIG. **8** shows a third pressure profile **142** that illustrates the additional effect of axially-aligned gravity on the axial pressure distribution **138** from FIG. **7** during operation. The static pressure bias from one end to the other is merely the equivalent hydrostatic head produced by gravity.

FIG. **9** shows a typical data store apparatus **144** adapted for use of the hydrodynamic bearing assembly **10** of this invention. A plurality of rotatable data storage disks, exemplified by disk **146**, is shown disposed for mounting on hub **108**, which is affixed to assembly **10** for rotation. A head assembly **148** is also shown disposed for moveable engagement with the surfaces of rotatable data storage disks **146** in the manner well-known in the art.

Clearly, other embodiments and modifications of this invention may occur readily to those of ordinary skill in the art in view of these teachings. Therefore, this invention is to be limited only by the following claims, which include all such embodiments and modifications when viewed in conjunction with the above specification and accompanying drawing.

I claim:

1. A hydrodynamic bearing assembly for supporting rotation of an object about a bearing axis, said assembly comprising:

a support sleeve having an inner sleeve diameter centered on said bearing axis and untapered at each of two sleeve ends and having one or more radial bearing zones and at least one axial thrust bearing zone with two axis-normal sleeve shoulders formed by steps in said inner sleeve diameter;

a bearing shaft having an outer shaft diameter centered on said bearing axis and tapered at each of two shaft ends and having one or more journals each disposed between two circumferential undercuts and having at least one thrust plate with two axis-normal plate faces formed by steps in said outer shaft diameter disposed between two circumferential undercuts, said bearing shaft being disposed rotatably within said support sleeve to form a substantially cylindrical radial-bearing clearance between each said journal and a corresponding said radial bearing zone and to form a substantially annular thrust-bearing clearance between each said axis-normal plate face and a corresponding said axis-normal sleeve and to form a tapered clearance between each said shaft end and a corresponding said sleeve end, said annular thrust-bearing clearances being coupled to one another by a non-bearing peripheral clearance between said thrust plate and said support sleeve, all said clearances being filled with a lubricating liquid that forms a surface-tension seal at each said tapered clearance, wherein a surface on at least one side of each said cylindrical radial-bearing clearance and a surface on at least one side of each said annular thrust-bearing clearance have surface-relief patterns each disposed to increase hydrodynamic pressure in the corresponding said clearance responsive to rotation of said bearing shaft with respect to said support sleeve; and

a plurality of passages within said bearing shaft each disposed to communicate fluid pressure between at least two said circumferential undercuts.

2. The hydrodynamic bearing assembly of claim **1** wherein said surface-relief pattern disposed on said at least one side of said each annular thrust-bearing clearance operates to urge a radially-outward flow of said lubricating liquid responsive to said rotation of said bearing shaft.

3. The hydrodynamic bearing assembly of claim **2** wherein said radially-outward flows of said lubricating liquid in said annular thrust-bearing clearances offset one another to create an elevated hydrostatic pressure in said non-bearing peripheral clearance without net fluid flow through said non-bearing peripheral clearance.

4. The hydrodynamic bearing assembly of claim **3** further comprising:

a central passage disposed within said bearing shaft to communicate fluid pressure among said plurality of passages.

5. The hydrodynamic bearing assembly of claim **4** further comprising:

a barrier-film coating on said sleeve ends for discouraging sleeve-end surface wetting by said lubricating liquid.

6. The hydrodynamic bearing of claim **5** comprising:

two adjacent said radial bearing zones disposed adjacent one said axial thrust bearing zone wherein all said surface-relief patterns are on said bearing shaft.

7. The hydrodynamic bearing of claim **1** comprising:

two adjacent said radial bearing zones disposed adjacent one said axial thrust bearing zone.

8. The hydrodynamic bearing assembly of claim **7** wherein said surface-relief pattern disposed on said at least one side of said each annular thrust-bearing clearance operates to urge a radially-outward flow of said lubricating liquid responsive to said rotation of said bearing shaft.

9. A hydrodynamic bearing assembly for supporting rotation of an object about a bearing axis, said assembly comprising:

a support sleeve having an inner sleeve diameter centered on said bearing axis and untapered at each of two sleeve ends and having one or more radial bearing zones each disposed between two circumferential undercuts and having at least one axial thrust bearing zone with two axis-normal sleeve shoulders formed by steps in said inner sleeve diameter;

a bearing shaft having an outer diameter centered on said bearing axis and tapered at each of two shaft ends and having one or more journals and having at least one thrust plate with two axis-normal plate faces formed by steps in said outer shaft diameter disposed between two circumferential undercuts, said bearing shaft being disposed rotatably within said support sleeve to form a substantially cylindrical radial-bearing clearance between each said journal and a corresponding said radial bearing zone and to form a substantially annular thrust-bearing clearance between each said axis-normal plate face and a corresponding said axis-normal sleeve shoulder and to form a tapered clearance between each said shaft end and a corresponding said sleeve end, said annular thrust-bearing clearances being coupled to one another by a non-bearing peripheral clearance between said thrust plate and said support sleeve, all said clearances being filled with a lubricating liquid that forms a surface tension seal at each said tapered

clearance, wherein a surface on at least one side of each said cylindrical radial-bearing clearance and a surface on at least one side of each said annular thrust-bearing clearance have surface-relief patterns each disposed to increase hydrodynamic pressure in the corresponding said clearance responsive to rotation of said bearing shaft with respect to said support sleeve; and

a plurality of passages within said bearing shaft each disposed to participate in communicating fluid pressure between said circumferential undercuts.

10. The hydrodynamic bearing assembly of claim **9** wherein said surface-relief pattern disposed on said at least one side of said each annular thrust-bearing clearance operates to urge a radially-outward flow of said lubricating liquid responsive to said rotation of said bearing shaft.

11. The hydrodynamic bearing assembly of claim **10** wherein said radially-outward flows of said lubricating liquid in said annular thrust-bearing clearances offset one another to create an elevated hydrostatic pressure in said non-bearing peripheral clearance without net fluid flow through said non-bearing peripheral clearance.

12. The hydrodynamic bearing assembly of claim **11** further comprising:

a central passage disposed within said bearing shaft to communicate fluid pressure among said plurality of passages.

13. The hydrodynamic bearing assembly of claim **12** further comprising:

a barrier-film coating on said sleeve ends for discouraging sleeve-end surface wetting by said lubricating liquid.

14. The hydrodynamic bearing assembly of claim **13** comprising:

two adjacent said radial bearing zones disposed adjacent one said axial thrust bearing zone wherein all said surface-relief patterns are on said bearing shaft.

15. The hydrodynamic bearing assembly of claim **9** comprising:

two adjacent said radial bearing zones disposed adjacent one said axial thrust bearing zone.

16. The hydrodynamic bearing assembly of claim **15** wherein said surface-relief pattern disposed on said at least one side of said each annular thrust-bearing clearance operates to urge a radially-outward flow of said lubricating liquid responsive to said rotation of said bearing shaft.

17. A data store apparatus for use with a host computer, said data store apparatus comprising:

a disk assembly having at least one rotatable data storage disk with at least one surface adapted for storage of data thereon;

a disk transport mechanism coupled to said rotatable data storage disk and including a motor for selectively imparting rotational motion to said data storage disk; and

a fluid bearing in said disk transport mechanism for supporting said rotatable data storage disk for rotation about a bearing axis, said fluid bearing including

a support sleeve having an inner sleeve diameter centered on said bearing axis and untapered at each of two sleeve ends and having one or more radial bearing zones and having at least one axial thrust bearing zone with two axis-normal sleeve shoulders formed by steps in said inner sleeve diameter,

a bearing shaft having an outer shaft diameter centered on said bearing axis and tapered at each of two shaft ends and having one or more journals each disposed between two circumferential undercuts and having at least one thrust plate with two axis-normal plate surfaces formed by steps in said outer shaft diameter disposed between two circumferential undercuts, said bearing shaft being disposed rotatably within said support sleeve to form a substantially cylindrical radial-bearing clearance between each said journal and a corresponding said radial bearing zone and to form a substantially annular thrust-bearing clearance between each said axis-normal plate face and a corresponding said axis-normal sleeve shoulder and to form a tapered clearance between each said shaft end and a corresponding said sleeve end, said annular thrust-bearing clearances being coupled to one another by a non-bearing peripheral clearance between said thrust plate and said support sleeve, all said clearances being filled with a lubricating liquid that forms a surface tension seal at each said tapered clearance, wherein a surface on at least one side of each said cylindrical radial-bearing clearance and a surface on at least one side of each said annular thrust-bearing clearance have surface-relief patterns each disposed to increase hydrodynamic pressure in the corresponding said clearance responsive to rotation of said bearing shaft with respect to said support sleeve, and

a plurality of passages within said bearing shaft each disposed to communicate fluid pressure between at least two said circumferential undercuts.

18. The data store apparatus of claim **17** wherein said surface-relief pattern disposed on said at least one side of said each annular thrust-bearing clearance operates to urge a radially-outward flow of said lubricating liquid responsive to said rotation of said bearing shaft.

19. The hydrodynamic bearing assembly of claim **18** wherein said radially-outward flows of said lubricating liquid in said annular thrust-bearing clearances offset one another to create an elevated hydrostatic pressure in said non-bearing peripheral clearance without net fluid flow through said non-bearing peripheral clearance.

20. The data store apparatus of claim **19** further comprising:

a central passage disposed within said bearing shaft to communicate fluid pressure among said plurality of passages.

21. The data store apparatus of claim **20** further comprising:

a barrier-film coating on said sleeve ends for discouraging sleeve-end surface wetting by said lubricating liquid.

22. The data store apparatus of claim **21** comprising:

two adjacent said radial bearing zones disposed adjacent one said axial thrust bearing zone wherein all said surface-relief patterns are on said bearing shaft.

23. The data store apparatus of claim **17** comprising:

two adjacent said radial bearing zones disposed adjacent one said axial thrust bearing zone.

24. The data store apparatus of claim **23** wherein said surface-relief pattern disposed on said at least one side of said each annular thrust-bearing clearance operates to urge a radially-outward flow of said lubricating liquid responsive to said rotation of said bearing shaft.

25. A data store apparatus for use with a host computer, said data store apparatus comprising:

a disk assembly having at least one rotatable data storage disk with at least one surface adapted for storage of data thereon;

a disk transport mechanism coupled to said rotatable data storage disk and including a motor for selectively

imparting rotational motion to said data storage disk; and

a fluid bearing in said disk transport mechanism for supporting said rotatable data storage disk for rotation about a bearing axis, said fluid bearing including

a support sleeve having an inner sleeve diameter centered on said bearing axis and untapered at each of two sleeve ends surfaces and having one or more radial bearing zones each disposed between two circumferential undercuts and having at least one axial thrust bearing zone with two axis-normal sleeve shoulders formed by steps in said inner sleeve diameter;

a bearing shaft having an outer shaft diameter centered on said bearing axis and tapered at each of two shaft ends and having one or more journals and having at least one thrust plate with two axis-normal plate faces formed by steps in said outer shaft diameter disposed between two circumferential undercuts, said bearing shaft being disposed rotatably within said support sleeve to form a substantially cylindrical radial-bearing clearance between each said journal and a corresponding said radial bearing zone and to form a substantially annular thrust-bearing clearance between each said axis-normal plate face and a corresponding said axis-normal sleeve shoulder and to form a tapered clearance between each said shaft end and a corresponding said sleeve end, said annular thrust-bearing clearances being coupled to one another by a non-bearing peripheral clearance between said thrust plate and said support sleeve, all said clearances being filled with a lubricating liquid that forms a surface tension seal at each said tapered clearance, wherein a surface on at least one side of each said cylindrical radial-bearing clearance and a surface on at least one side of each said annular thrust-bearing clearance have surface-relief patterns each disposed to increase hydrodynamic pressure in the corresponding said clearance responsive to rotation of said bearing shaft with respect to said support sleeve; and

a plurality of passages within said bearing shaft each disposed to participate in communicating fluid pressure between said circumferential undercuts.

26. The data store apparatus of claim **25** wherein said surface-relief pattern disposed on said at least one side of said each annular thrust-bearing clearance operates to urge a radially-outward flow of said lubricating liquid responsive to said rotation of said bearing shaft.

27. The hydrodynamic bearing assembly of claim **26** wherein said radially-outward flows of said lubricating liquid in said annular thrust-bearing clearances offset one another to create an elevated hydrostatic pressure in said non-bearing peripheral clearance without net fluid flow through said non-bearing peripheral clearance.

28. The data store apparatus of claim **27** further comprising:

a central passage disposed to communicate fluid pressure among said plurality of passages.

29. The data store apparatus of claim **28** further comprising:

a barrier-film coating on said sleeve ends for discouraging sleeve-end surface wetting by said lubricating liquid.

30. The data store apparatus of claim **29** comprising:

two adjacent said radial bearing zones disposed adjacent one said axial thrust bearing zone wherein all said surface-relief patterns are on said bearing shaft.

31. The data store apparatus of claim **25** comprising:

two adjacent said radial bearing zones disposed adjacent one said axial thrust bearing zone.

32. The data store apparatus of claim **31** wherein said surface-relief pattern disposed on said at least one side of said each annular thrust-bearing clearance operates to urge a radially-outward flow of said lubricating liquid responsive to said rotation of said bearing shaft.

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